S·A·E JOURNAL

PUBLISHED BY THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.

D. G. Roos, President — David Beecroft, Treasurer

John A. C. Warner, Secretary and General Manager

Norman G. Shidle, Executive Editor

Vol. 35 — OCTOBER, 1934 — No. 4

CONTENTS

Engine Types Adapted To Car Design Trends—L. P. Kalb 13

Production Men! *Your* Meeting Is Ready for Oct. 10 18

News of the Society 20

S. A. E. Nominees for 1935 21

What Members Are Doing 23

Meetings Calendar 24

Transactions Section Begins

Chemical Hay for Mechanical Horses—Robert E. Wilson and D. P. Barnard 359

Fin and Cylinder-Baffle Design for Air-Cooled Engines—Carlton Kemper . 370

A Suggested Index of Diesel Fuel Performance—A. E. Becker and H. G. M. Fischer 376

The Hesselman Low - Compression Diesel-Fuel-Burning Engine—A. W. Pope, Jr. 385

Transactions Section Ends

New Members Qualified 25

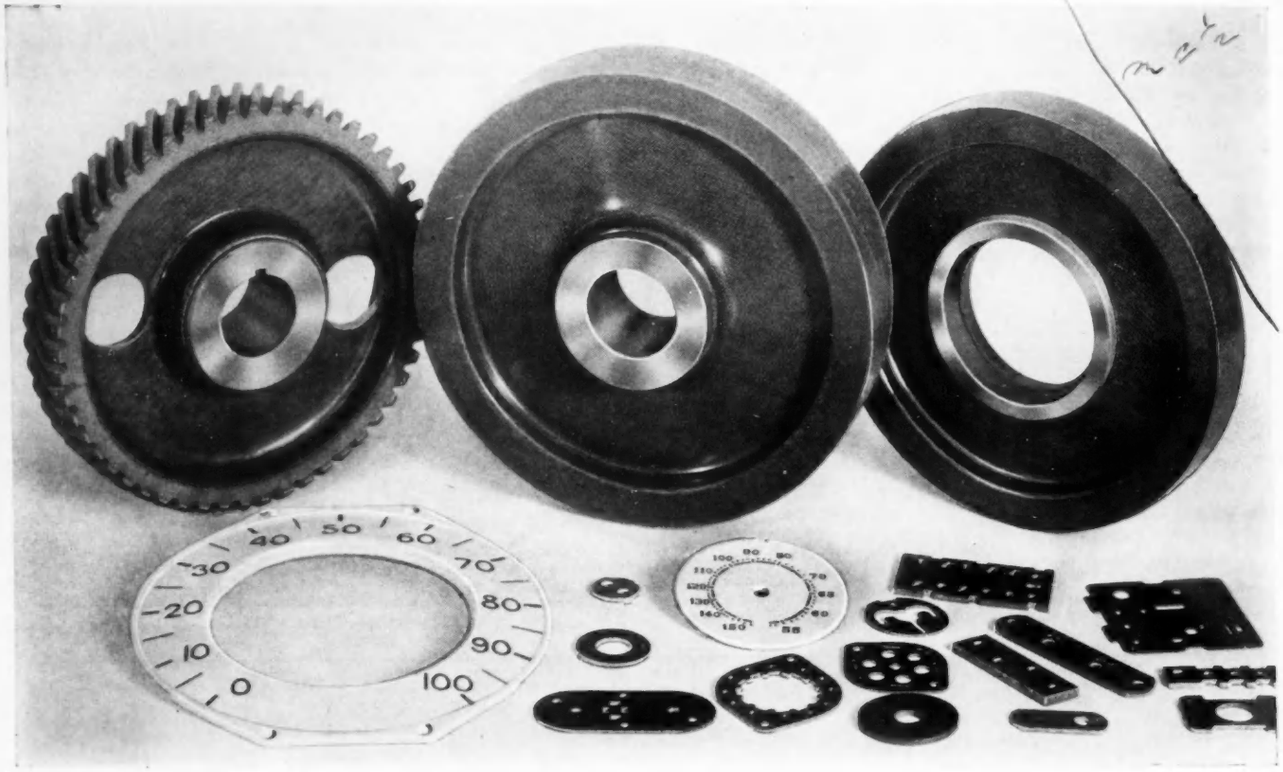
Applications Received 25

Behind the Scenes with the Committees 30

Notes and Reviews 32

Publication Office, 56th and Chestnut Sts., Philadelphia, Pa.; Editorial and Advertising Departments at the headquarters of the Society, 29 West 39th St., New York, N. Y. Western Advertising Office, Room 2-136 General Motors Bldg., Detroit, Mich.

The Society is not responsible for statements or opinions advanced in papers or discussions at its meetings or in articles in the JOURNAL.



INSUROK

(A Superior Laminated Phenolic Product)

From a tiny washer to a gigantic gear you will find a molded or laminated Insurok to meet your requirements.

INSUROK is available in a multitude of forms in a grade and thickness for every service . . for high heat resistance . . for unusual toughness . . for severe mechanical service . . for high dielectric, as well as a wide range of colors and finishes to suit varying applications.

The Richardson organization, schooled in cooperating with the car manufacturer and conscious of the plus type of service required, are always at your command.

Why INSUROK replaces other materials

Tougher!
More resilient!
Uniform in structure!
Easily fabricated!
Stronger for its weight!
High dielectric!
Non-conductor of heat!
Non-corrosive!
Mechanical shock absorber!
Fabricated to your order!

The RICHARDSON COMPANY

MELROSE PARK, ILL. NEW BRUNSWICK, N. J. CINCINNATI, OHIO INDIANAPOLIS, IND.

DETROIT OFFICE 4-252 G. M. Bldg., Phone Madison 9386

CUSTOM MOLDERS OF: INSUROK (Phenolic Resin) . . . EBROK (Bituminous Plastic) . . . RUBTEX (Hard Rubber) . . . RICHELAIN (Urea Resins)

Engine Types Adapted To Car Design Trends

By L. P. Kalb
Continental Motors Corp.

THE American automobile industry has adhered to the in-line and V-type engines exclusively for so many years that it is doubtful if the majority of our engineers have given any consideration whatever to other engine arrangements. Of course there has been very little incentive for them to do so, as these conventional types of engine have met the requirements of the automobile in a very satisfactory manner. Their suitability to the structure and arrangement of the automobile, as we have known it, is due primarily to the fact that engines and vehicles have grown up together. In fact, the evolution of all multi-cylinder engines has been influenced largely by the necessity for adaptation to the vehicles or machines in which they are used. In this respect there exists a close analogy between mechanical and biological evolution. This process of mechanical adaptation to environment is most strikingly exemplified by the wide divergence of automobile and aircraft engines.

A study of automobile history shows the strong influence that car structure and engine arrangement have had on each other. Reviewing the cars displayed at the New York Shows in the transition years of 1904, 1905 and 1906, it is plainly seen that the change of engine location from underneath to in front of the passengers went hand in hand with the evolution from horizontal one and two-cylinder to vertical four and six-cylinder engines.

Since that time, progress in engine design has consisted mainly of increase in power, part of which has been accomplished by improvement of specific output, but the large part has been due to increasing the number of cylinders. The only basic change in engine arrangement that has been made was the addition of the V-type engine. At the present time the same spirit of change seems to be creeping into automobile design that has pervaded the social and economic life of the world. Almost every make of car today has some individual, if not entirely novel, feature. With radical changes in car structure in the offing, it is probable that changes in engine arrangement are also imminent.

The purpose of this paper is to discuss unusual forms of engines, and compare their adaptability to the newer types of automobile with the conventional engines now in general use. Discussion will be confined to those for which there is some background of successful application, if not in automobile practice, then in some kindred automotive field. The types that are to be discussed, and the reasons for including them, are as follows:

(1) The in-line engine, because it is an accepted standard of the automobile industry.

(2) The V-type engine, for a similar reason.

(3) The flat engine, because of its history in the automobile industry, and because it offers attractive possibilities for certain car structures.

(4) The barrel engine, because of its many interesting features and the tremendous amount of thought and effort exerted on it by inventors and engineers.

(5) The radial engine, because of its general use in the aeroplane and because of the possibilities which it offers for adaptation to the automobile, particularly one with the engine in the rear.

THIS article discusses unusual forms of engines and compares their adaptability to the newer types of automobile with the conventional engines now in general use. Discussion has been confined to those engines for which there is some background of successful application, either in the automobile or in some kindred automotive field. The types discussed include: The in-line engine, the V-type engine, the flat engine, the barrel engine and the radial engine.

The advantages and disadvantages of these various engines are discussed with respect to ease of adaptation to various types of cars, balance, even firing, torsional conditions, weight, compactness, ease of manifolding, accessibility and ease of manufacture. It is brought out that the 4-cylinder opposed engine has many attractive features which would lend themselves to a low-cost automobile. The attractive possibilities of adapting the radial engine to the rear-engine type of car also are pointed out.

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

In discussing the adaptability of the above engines, it will be interesting to compare them with respect to balance possibilities, firing intervals, susceptibility to torsional vibration, manifolding, accessibility, weight, space requirements and manufacturing cost. Efficiency, specific output and other characteristics of this nature have no place in this discussion as they are dependent on internal design and not on engine arrangement. To assist in this comparison there is included, as an appendix, an analysis, from the standpoint of balance and firing intervals, of a large number of cylinder arrangements using the crank type of mechanism. This was prepared by A. J. Meyer of Continental Motors Corp.

The In-Line Engine

The history or background of application of this type of engine in the automobile field needs very little amplification. The fact that it is the preferred, if not the generally accepted type, proves its suitability to the automobile, as we have known it, throughout all of the industry's mature years. Its evolution was a logical one, consisting merely of the multiplication in series of a previously existing mechanism. Increasing piston displacement, and consequently engine output, in this manner, not only progressively improves the smoothness of torque application, but also improves inertia balance. As more cylinders are added, dynamic balance is improved until when six cylinders are reached we arrive at a condition of inherent cancellation of inertia effects. Increasing the number of cylinders above this critical number cannot further improve the condition of balance, but the torque variation at low speed is appreciably reduced in the eight, as compared with the six.

Accessibility and ease of manufacture are considered two of the line engine's outstanding advantages. It is doubtful if the latter advantage is inherent in this form of engine, but it is due rather to the perfection of manufacturing processes and the development of specialized machinery, which high volume has made possible. Without our present day manufacturing facilities, I doubt that anyone would consider an enbloc eight-cylinder engine easy to produce, either from the foundry or machine shop standpoint.

It must be admitted that the in-line engine possesses certain inherent disadvantages, and one of the most important of these is the fact that this arrangement of cylinders undoubtedly produces the heaviest possible engine for a given displacement. Practical observation will disclose this fact, and it can also be demonstrated by simple reasoning, at least for the crankcase part of the structure. Since in high-speed engines, stiffness considerations govern the design, we will use for our illustration the deflection formula for a rectangular beam in its most general form:

$$f = \frac{K\,PL^3}{bh^3}$$

where

f = Deflection of the beam in the place of load application
K = A constant, depending only on method of loading and supporting
P = Transverse load on the beam
b = Width of beam
h = Height of beam
l = Length of beam

Thus, since we know that for a given stroke the height of the engine will remain constant for any number of cylinders, it can be seen that for a given load and deflection the section width, and, consequently, the section weight, will increase as the cube of the length. Therefore the beam weight is proportional to the fourth power of the length, and, since the power increases directly as the length, it follows that the weight per horsepower will be proportional to the cube of the length. This method of illustrating the point is too much of a generality to be entirely convincing, but if the expression for deflection be written to incorporate the actual moment of inertia of an engine crankcase, the result will be practically the same.

Another disadvantage of the line type engine is its susceptibility to torsional vibrations. As dynamic balance is improved by the multiplication of cylinders, torsional roughness becomes worse, unless the stiffness of the shaft is increased to compensate for its increased length. Increasing the shaft diameter, however, has an undesirable effect on rubbing factors, friction and temperature, and again is responsible for an increase in weight per unit of power.

Manifolding is also complicated in the line engine by the unequal distances that the gas must travel from carbureter to cylinders. Here again, the greater the number of cylinders, the worse this problem becomes. These unequal distances produce uneven resistance to flow and uneven ramming effect, making it difficult to maintain equal filling of all cylinders. The unequal surface areas cause variation in evaporation in the manifold, which in turn makes it difficult to deliver equal quantities of fuel to all cylinders.

None of the above disadvantages have in the past proven of sufficient consequence to outweigh the engine's many advantages. It is only when we come to apply this type to some new form of automobile that the better adaptability of some other arrangement may throw the balance in its favor.

The length of the in-line eight has always been more or less of a handicap to car installation. It becomes a serious obstacle to moving passenger space forward, and certainly does not contribute to a satisfactory installation in the rear of the car. Those of you who saw Sir Denniston Burney's car at the 1932 Annual Meeting of the Society must have been impressed by this fact. Builders of cars with line engines in the rear have tried to make a virtue out of necessity by incorporating a fish-tail in the rear of the body, for which they claim certain aerodynamic justification. It is doubtful if such advantages offset the blocking off of rear vision and the increase in car length and weight which this entails. Surely the same benefits can be obtained by some simpler means, such as a slot or flap, which would have a less detrimental effect on maneuverability. Most automobiles have to be backed or parked much more frequently than they are driven at speeds requiring the last word in streamlining.

Mounting the engine in the rear of the car has many obvious advantages, not the least of which is the fact that with the engine in this position its noise, fumes and heat are less apparent to the passengers and consequently less detrimental to their comfort. With the conventional line engine, however, it is doubtful if the advantages of the rear location would offset its disadvantages.

V-Type Engines

Although the use of this form of engine has never been as general as that of the in-line type, the character of the cars in which it has been used, as well as its long history of successful service, can leave no doubt as to its adaptability to the automobile.

One of its outstanding advantages, when compared with

the line engine, is the fact that it lends itself to the crowding of a greater amount of power into a given hood space, and, consequently, into a given wheelbase. It is also somewhat lighter than a line engine of the same displacement and output.

The fundamental balance possibilities of the V engine are rather attractive. If we analyze the simplest form of V engine, i.e., two cylinders at 90 deg., operating on one crank, we find a condition where unbalanced primary inertia forces are constant and independent of crank angle. Thus primary unbalance can be cancelled out by properly placed counterweights. Consequently, since any V engine can be made up of a number of these 90 deg. two-cylinder units, perfect, though not inherent, primary balance can be attained. The balance of the V engine is inferior to that of the line engine in that inherent cancellation of secondary inertia forces cannot be attained with fewer than eight cylinders. Furthermore, although the V-8 with a 90 deg. crankshaft does possess perfect balance and even firing intervals, there exists a rather unfavorable condition in regard to firing sequence, which complicates the manifold problem. This is due to the firing of two adjacent cylinders within 90 deg. of each other, which makes it undesirable to use "siamesed" ports. The 1934 Ford gets around this by the use of separate ports for each cylinder, which, in the writer's estimation, is the proper way to manifold any engine. The V-8 is, of course, superior to the line eight from the standpoint of torsional balance, due to its reduced crankshaft length, although a good deal of this advantage must be sacrificed again due to the necessity of using heavy counterweights.

The one respect in which the V-type engine compares most unfavorably with the line type is that of accessibility.

From the standpoint of cost the V-8 is probably about on a par with the line-8, the saving in weight, and consequently material cost, being offset by an increase in machining and assembly cost. The fact that it is possible to build an inherently balanced in-line engine with a smaller number of cylinders than with the V arrangement throws the cost comparison somewhat in favor of the former, when considered in the light of the low-cost automobile.

The lower weight and reduced length of the V-8 are worthwhile assets, for application either to a car with passenger space moved forward or with the engine in the rear. A good example illustrating the adaptability of this engine to the rear of an automobile is the Tatra car shown in Fig. 4.

The V-12 and V-16 both consist of two inherently balanced engines, and consequently must be in perfect balance themselves. Both possess exceedingly smooth torque diagrams. Neither a line 12 nor 16 would be practical for automobile use for the reason of extreme length and excessive torsional vibration. These reasons, together with other advantages inherent in V-type engines, account for the V-12 and V-16 enjoying the preponderance of favor in the *de luxe* automobile field.

The Flat or Horizontal Engine

This classification, naturally, includes both the line engine lying on its side and the horizontal opposed engine. There is considerable background of automobile use for both. In fact, it is fairly obvious that had the automobile engine remained under the car, as was the practice in most of the primitive models, the horizontal arrangement, instead of the vertical, would today be the accepted standard.

In bus and truck practice, there seems to be, at present, a revival of the attempt to get the engine away from valuable load and passenger carrying space. Apparently it is the belief of many engineers that engine reliability has reached the stage that accessibility can be sacrificed. This trend, of course, is being helped along by legislation limiting overall vehicle dimensions. The adaptability of the flat engine to this form of vehicle is evidenced by its recent adoption by two of our leading manufacturers. The flat engine would undoubtedly lend itself to the "Airflow" type of automobile.

With the wide hood space available, increased engine width is not objectionable, while with the hood opening provided, accessibility would be improved as far as points requiring comparatively frequent attention are concerned. This applies to tappets, water pump, electrical equipment, etc. The difficulty of grinding valves, removing pistons, and honing cylinders would be increased, but as the necessity for these operations is comparatively much less frequent, it is apparent that the accessibility score is decidedly in favor of the flat engine. Surely anything that would eliminate the necessity for removing a wheel in order to adjust tappets would be a step in the right direction.

Moving the passenger-carrying space forward makes a reduction of engine length desirable. From this standpoint the opposed engine is even better adapted to this type of car than the horizontal line engine.

When we compare the balance possibilities of the opposed and line engines we find a rather mixed condition. Where a small number of cylinders is used, the opposed engines give better balance than can be obtained with the in-line arrangement, but to obtain perfect inherent balance we require at least 12 cylinders, as compared with six in the line engine and eight in the V-type. The analysis of one of the simplest forms of opposed engine, i.e., two cylinders operating on cranks 180 deg. apart, reveals cancellation of both primary and secondary inertia forces in the transverse plane. Primary and secondary rocking couples remain, however, that cannot be cancelled by counterweights. As we increase the number of these two-cylinder units, it is possible to inherently cancel out these rocking couples, but not until twelve cylinders are reached can perfect balance, together with equal firing intervals, be attained.

The four-cylinder opposed engine with a flat four-throw crankshaft, looking-glass symmetry, No. 1 and No. 3 cylinders on one side and No. 2 and No. 4 on the other, is in primary and secondary balance, has equal firing intervals, but possesses a secondary rocking couple, which however is quite small. Perfect dynamic balance could be achieved in this engine by placing No. 1 and No. 4 cylinders on one side and No. 2 and No. 3 on the other. However this would produce the undesirable effect of two cylinders firing simultaneously at 360-deg. intervals instead of 180, as in the first arrangement. The resultant effect on the torque diagram would be more undesirable than the unbalanced secondary rocking couple remaining in the first arrangement.

A six-cylinder opposed engine can hardly be considered attractive, particularly from the balance standpoint. With a three-throw crankshaft unequal firing intervals are unavoidable. Although the transverse plane components of both primary and secondary forces cancel out, both primary and secondary rocking couples remain. With a six-throw crankshaft unsymmetrical spacing of cranks would be required to produce even firing intervals, but, of course, a detrimental effect on balance would result. Any form of opposed six, consequently, would compare quite unfavorably with the perfectly balanced in-line six.

An eight-cylinder opposed engine, with a flat four-throw crankshaft, can be in both primary and secondary balance,

with rocking couples cancelled out. This type of engine would again possess the disadvantage of two cylinders firing simultaneously, with its resultant undesirable effect on the torque diagram. This condition cannot be overcome without disturbing balance. An opposed eight with a 90-deg., four-throw crankshaft would not possess the same good balance as the 90 deg. V with a 90 deg. crank. Even firing intervals could be maintained and secondary forces cancelled out, but a primary rocking couple of considerable magnitude located in the horizontal plane would remain. This could not be counterbalanced without introducing a couple of equal severity in the vertical plane.

Since the opposed twelve also consists of two inherently balanced six-cylinder units, there can be no advantage or disadvantage from the balance standpoint over the V-type. Firing occurs at equal intervals of 60 deg., the same as in the V. Consequently any superiority that the opposed twelve could have over the V-type twelve would be a structural one or a better adaptation to space requirements.

An opposed sixteen would possess equal balance qualities with the V-16, but would have the undesirable feature of simultaneous firing of two cylinders. The 135 deg. V is preferable from this standpoint.

It is the writer's belief that the opposed engine with most attractive possibilities for automobile use is the four cylinder. This would lend itself exceedingly well to a low priced car with either front or rear engine location. Compared to the line four, which has in the past dominated the low priced automobile field, the opposed four would possess the following advantages: It would occupy less space, the balance would be better and manifolding should be an easier problem. Taking everything into consideration this engine has much to offer the low priced automobile. Again a Tatra model can be cited as a worthy example of the application of this type of engine.

The Barrel Engine

In this classification are included all engines with the cylinders parallel to the shaft. Although possessing no background of history for automobile use, the amount of effort that has been expended on engines falling in this category is rather surprising. The paper by E. S. Hall, read before the Indiana Section of the Society in November, 1930, gives a remarkably comprehensive description of the various engines of this type that had been built. Mr. Hall was engaged for a number of years in the design and development of the Michell Swash Plate Engine, and is, I believe, the outstanding authority on this subject. In view of this I will not devote any time to discussion of the various engines in this classification as I am sure there is nothing that I could add to Mr. Hall's paper.

In spite of the many barrel engines that have been worked on from the development standpoint, the only one that ever reached the production stage is the swash-plate marine engine announced by the Sterling Engine Co. last year.

The most attractive feature of this type of engine is its compactness. This feature, of course, lends itself both to moving passenger space forward, or to rear engine application. This compactness also makes for low weight and should, in the end, make for low cost. From the standpoint of balance, the swash-plate engine is superior to all other arrangements in that perfect balance can be obtained with as few as three cylinders. The most serious disadvantage is that of accessibility. There is no denying that this form of engine is very attractive and is well worth the consideration of engineers for application to the automobile, particularly to the advanced forms, which have provided the excuse for writing this paper.

The Radial Engine

Although never applied to the automobile, the use of this type of engine in aeroplanes has become so general that other types can be considered as rare exceptions. The radial has two outstanding advantages over other types, both of which render it desirable for aeronautical applications. The first and most important of these is its low weight, which, of course, is a very desirable feature for any automotive use. The reason for this low weight is due, primarily, to its multiple arrangement of cylinders around a compact crankshaft and crankcase, and not entirely due to the use of light alloys, as may be the general belief. Whereas five to six pounds per horsepower is considered good practice in the in-line automobile engine, aeronautical engineers are not satisfied with 2 lb. per hp., and present trends would indicate that within a short time, 1 lb. per hp. will be considered ordinary practice. It is estimated that in a radial engine for use in an automobile, 3 to 3½ lb. per hp. could be attained without departing from conventional automobile engine materials.

The second inherent advantage of the radial engine, from the aeronautical standpoint, is the fact that this cylinder arrangement lends itself to air cooling. This may not at present be considered an asset for automobile application, but, at least, with this cylinder arrangement, the undesirable features of air cooling an automobile engine would be reduced.

Probably the greatest advantage of the radial engine for automobile application would lie in its compactness and structural arrangement, both of which adapt it particularly well for a rear-engine installation. Compactness and low weight both tend to eliminate one of the particular disadvantages of the rear power plant, which is the overhanging weight behind the rear axle. Since both the length of the lever arm and the weight on the end of it are reduced, the undesirable effect of overhang is considerably lessened. (See Figs. 1 and 2.)

Automobile engineers with whom I have discussed this subject have offered as objections to the radial engine its comparatively low speed, its roughness and its noise. None of these defects are inherent in this type of engine. When we take into consideration the large displacement per cylinder of most radial aeroplane engines, their operating speed is actually quite high when compared with that of in-line automotive engines of the same size.

Although the speed of aeroplane engines has been purposely limited to correspond with efficient propeller speeds, there are several examples of geared down small radial engines whose operation speeds will compare with so-called high speed line engines. The push rod type of valve actuating mechanism required by the overhead air cooled cylinder is bound to be heavy and is in itself a speed limiting factor. However, when the radial engine is applied to the automobile, there is no reason why this type of valve mechanism should be retained. With a water-cooled cylinder the L-head or sleeve valve construction are just as applicable to the radial as to the in-line engine.

In regard to the roughness objection, few engineers unfamiliar with aircraft practice realize how good the balance of a radial engine can really be made. With five or more cylinders all centrifugal and inertia forces can be cancelled out by the proper design of master rod and proportionate counterweighting. Some small unbalance will remain if the

articulated rod construction is used. However this effect can also be completely taken care of with no more complication than the addition of a single rotating mass driven at twice crank speed. Even inherent cancellation of secondary forces is possible of attainment, although the means employed to do this do not have the background of experience possessed by the more conventional articulated rod.

In single sleeve valve engines even the motion of the valves is in complete inherent balance. It is quite probable that even the roughest radial aircraft engine would seem comparatively smooth, if the mass of the engine bore the same relationship to the force of the power impulse that it does in the automobile line engine, and if, in addition, it were as substantially mounted. It is hardly necessary to mention that, notwithstanding its flimsy crankshaft, torsional vibration in the radial engine proper (for example, without a propeller attached to it) is much less of a problem than in a line engine of the same number of cylinders.

Excessive noise is another objection that does not necessarily apply to the radial engine when used in an automobile. In fact, most of the noise generally attributed to the engine is not engine noise at all, but is caused by the propeller. This becomes fairly evident when one visits the propeller testing department at Wright Field. Although the propellers are driven by electric motors, the noise is not perceptibly less than when driven by engines. Any excessive noise that may rightfully be attributed to the aircraft radial engine is due to the type of valve gear commonly used, to the loose piston fits necessary with air cooling and to the gear trains employed.

With a water cooled cylinder, L-head or sleeve valve construction, and with a piston such as is commonly used in automobile practice, the greater part of these noises can be overcome. The gearing necessary for the actuation of the valves and accessories in a radial engine can also be made considerably quieter than is now done in aeronautical practice. The use of internal gears is quite feasible and particularly applicable to this type of engine. With the engine mounted in the rear of the car, the engine noise problem should not be as troublesome as it has been in the forward location, no matter what type of engine is used.

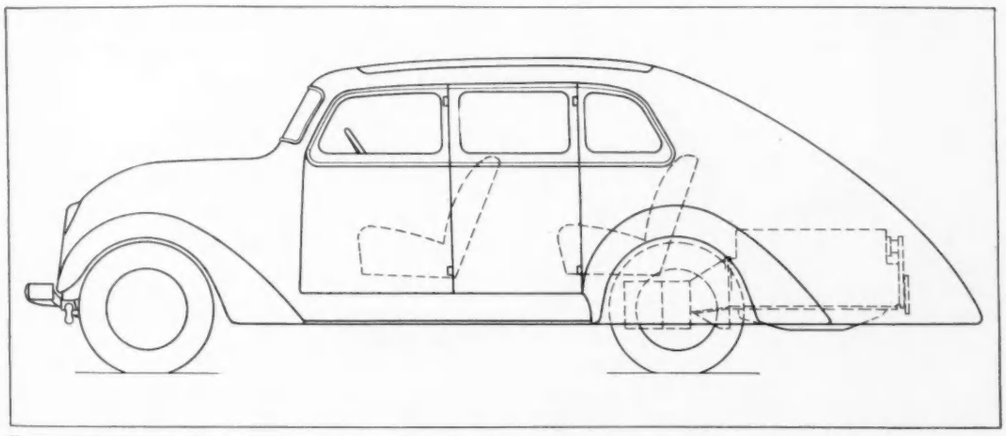
Another objection which has been raised is the fact that the radial aircraft engine does not operate satisfactorily at low speeds. This again is not inherent, but is due to the fact that in its aeronautical application, other considerations are sacrificed to high specific output at definite operating speeds. The writer is familiar with an installation of a radial engine in a ground vehicle in which idling and low speed torque has been made quite as good as that in line engines, without sacrificing anything at the top end.

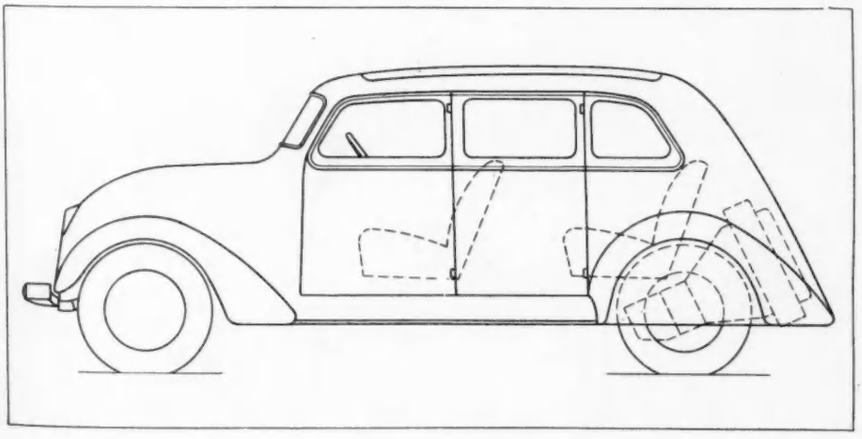
The radial arrangement actually lends itself to better manifolding than the line arrangement. In the so-called spider-web manifold used in the English Pobjoy engine the carbureter is mounted on the axial center of the engine and from a central distributing chamber the gas passages extend radially to all of the cylinders. This construction lends itself to the equal filling of all cylinders, as well as to uniform vaporization and delivery of fuel. The distance the gas travels from the distributing chamber to each cylinder being the same in each case, the possibilities for good filling, due to the development of high ram, are very favorable.

If the trend toward superchargers for automobile engines develops further, this will provide another advantage for the radial engine. The driving of the rotor can be accomplished in a more compact and generally satisfactory manner and the location of the blower on the axial center of the engine lends itself to a more efficient distribution of the gas mixture.

When it comes to production cost, I believe that with some study and develop-

(Cont. on page 26)





Figs. 1 (above) and 2 show how much might be gained in compactness by the use of a radial engine instead of a conventional in-line engine, where the passenger capacity of the two chassis is the same

Production Men! *Your*

PROCEEDING with the precision symbolized by the figure which appears at the left of this column, arrangements for the general Production Meeting of the Society in Detroit, Oct. 10 and 11, have advanced to the point where as this issue of the JOURNAL goes to press they can be presented in virtually complete form. Copies of each of the four papers have already been mimeographed and are ready for distribution to those who plan to attend the meeting.

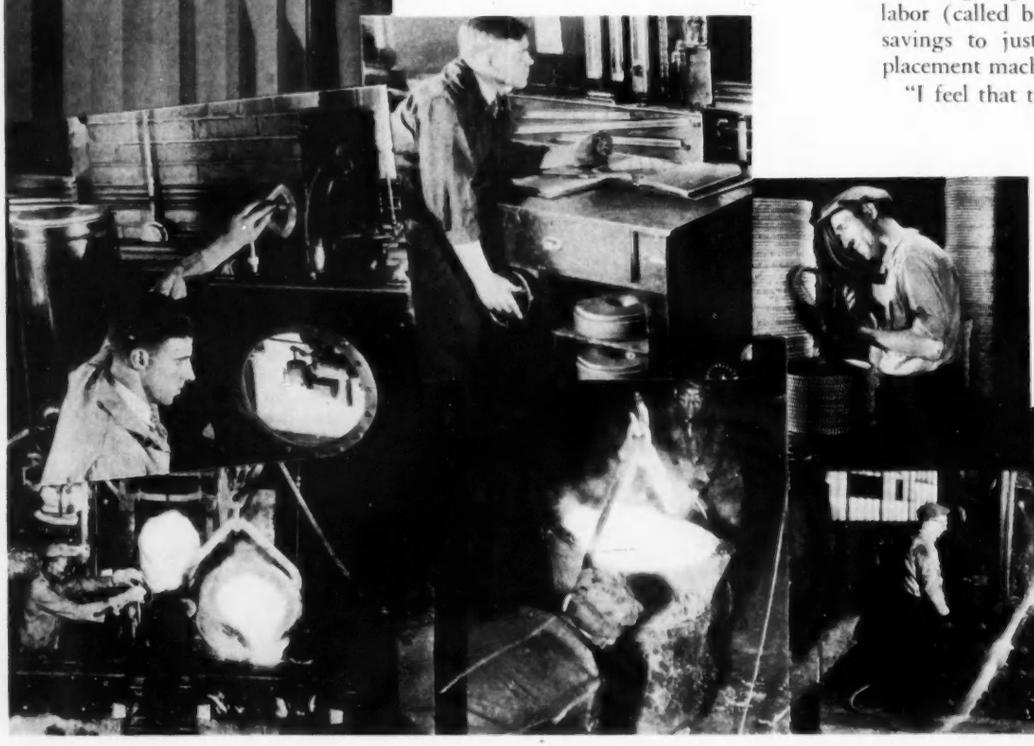
Considering that there have been no general Production Meetings during the last two years, an unusually large attendance is expected, this point of view being reinforced by the excellent quality of the papers for presentation. The first session scheduled, that on cutting oils, will treat broadly a subject which has been prominent in the minds of production men for a number of years. The paper by W. B. Huffman of the Chevrolet Gear and Axle Plant will cover the subject broadly with the collaboration of C. B. Harding of the Sun Oil Co., and W. H. Oldacre, D. A. Stuart & Co., Ltd. Some of the topics covered in the paper on "Cutting Oils and the Fundamental Characteristics in Their Selection," are types of cutting oils, non-emulsifying laboratory control and applications of the various types.

The paper by E. S. Chapman of the Amplex Division of Chrysler Motors, on "Production Experience with Surface Broaching" will bring up to date the review of the subject given by Joseph Geschelin of *Automotive Industries* at the last Annual Meeting of the Society in Detroit. Mr. Chapman's paper is extremely practical, being devoted almost entirely to descriptions of material, operations and equipment entering into the surface broaching operations.

In presenting his paper on "Machinery and Equipment Policies in View of the Present Business Situation," J. E. Padgett, Spicer Manufacturing Corp., says:

"I have been asked to prepare a paper outlining the effect on product cost resulting from the purchasing of replacement or additional equipment. This request came as a result of a discussion between a number of us at the Chicago Meeting a year ago, when we got into an argument concerning the question of including a percentage added to direct labor (called burden) when computing savings to justify the purchase of replacement machinery.

"I feel that this subject is entirely too



Photographs on these pages from General Motors Corp., E. G. Budd Mfg. Co., Chevrolet Motor Co., Bellanca Aircraft Corp., and (top right) Ewing Galloway.

Meeting Is Ready for Oct. 10

limited in its scope to be particularly useful or interesting to a group of this kind. I will go into a discussion of it briefly but will add, as the major part of the paper, some thoughts on general equipment policies which will probably undergo a very radical change in the future if the present legislative and financial conditions continue. These thoughts are entirely personal and are my reactions to what I have seen, read and discussed in the past several months."

The general interest and timeliness of Mr. Padgett's paper can be judged from the two paragraphs quoted above.

"Balancing Problems in Automotive Engineering" have been studied by T. C. Van Degrift and J. M. Tyler at the General Motors Research Laboratories, and their experience and conclusions will be recorded at the meeting.

Chairmen for each session have been carefully picked for their special interest in the subject to be treated.

It is regretted at this time that definite announcement cannot be made of the speaker for the Production Dinner. Every effort is being made to obtain a speaker of prominence, but difficulties in coordinating the date of the meeting with the plans of two or three speakers who may be available have held up the arrangements to date. The Detroit Section is cooperating actively in arrangements for the Dinner.

Program of S.A.E. Production Meeting, Oct. 10-11

Book-Cadillac Hotel — **Detroit**

WEDNESDAY, OCTOBER 10

Cutting-Oil Session—Afternoon
A. L. Clayden, *Chairman*

Cutting Lubricants and the Fundamental Characteristics in Their Selection—W. B. Huffman, chief chemist, Chevrolet Gear and Axle Plant, Detroit.

Collaborators—C. B. Harding, Sun Oil Co., Philadelphia, and W. H. Oldacre, D. A. Stuart & Co., Ltd., Chicago.

Dynamic Balancing Session—Evening
W. H. McCoy, *Chairman*

Balancing Problems in Automotive Engineering—T. C. Van Degrift and J. M. Tyler, General Motors Research Laboratories, Detroit.

THURSDAY, OCTOBER 11

Broaching Practice Session—Afternoon
Joseph Geschelin, *Chairman*

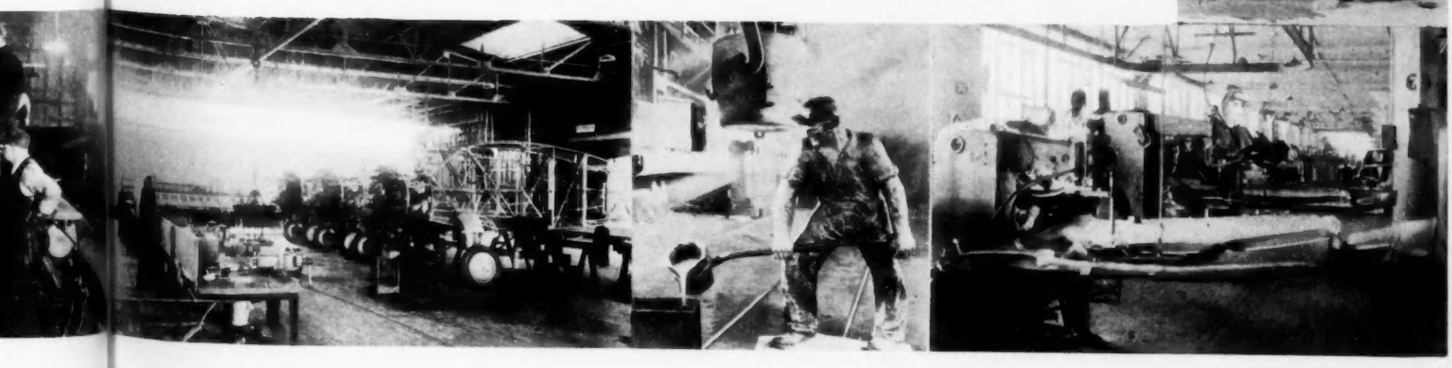
Production Experience with Surface Broaching—E. S. Chapman, Chrysler Motors, Amplex Division, Detroit.

Machinery and Equipment Policies in View of the Present Business Situation—J. E. Padgett, Spicer Mfg. Corp., Toledo, Ohio.

Dinner

Sponsored jointly by Detroit Section.

Toastmaster—Clyde R. Paton, Chairman, Detroit Section. Principal Speaker—R. H. Grant, vice-president, General Motors Corp.



News of the Society

Met. Section a Sponsor For Big T.&M. Meeting

SPONSORED jointly by the Metropolitan Section, S.A.E.; the New Jersey Motor Truck Association, and the Newark Chamber of Commerce, a three-day Regional Transportation and Maintenance Meeting will be held in Newark, N. J., Nov. 8 to 10. Morning, afternoon and evening sessions will be held on the first two days of the meeting, and the third will be given over to an inspection of the Newark Airport, and an exhibit of automotive equipment.

Almost every general subject of interest to transportation and maintenance men has been included on the program, which bristles with names of speakers nationally and internationally known in the field.

N. Mitchell of the Asiatic Petroleum Co., London, is coming over to present complete data on one of the largest Diesel-engine operations in the world, that of the London General Omnibus Co. The legal administration of highways will be represented by Harold G. Hoffman, motor vehicle commissioner of New Jersey. Other speakers are also of eminence in their vocations.

A tentative program follows:

THURSDAY, NOV. 8

Morning Session: Chairman—F. C. Horner, assistant to vice-president, General Motors Corp.

Opening Address—Senator J. G. Wolber.

Remarks—S. G. Harris, chairman, Metropolitan Section, S.A.E.

J. F. Winchester, president, New Jersey Motor Truck Association.

E. W. Wollmuth, executive vice-president, Newark Chamber of Commerce.

Taxation and Regulation as Applied to Highway Transportation—Discussion of Present Trends and Imminent Changes.

Speakers: Major R. F. Britton, director, National Highway Users Conference.

Gen. B. H. Markham, director, American Petroleum Industries Committee.

Afternoon Session: Chairman—T. B. Rendel, Shell Petroleum Co.

Automotive Diesel Operating Data—Records of the London General Omnibus Company.

Speaker: N. Mitchell, Asiatic Petroleum Co., London.

Evening Session: Chairman—Robert Jackson, chairman of New Jersey State Code Authority for the Trucking Industry.

Operation of the Code in the Trucking Industry.

Speaker: T. V. Rodgers, president of the American Trucking Association.

Economical Application of Rates for Movement of Freight by Motor Trucks.

Speaker: F. I. Hardy, industrial engineer, Boston.

FRIDAY, NOV. 9

Morning Session: Chairman—W. E. John.

The New Automotive Fuels—Butane and Propane as Refrigerants and Fuels.

Speaker: Open.

Moving Pictures of Fuel Tests in 1934 Cars on Uniontown Hill. Impartial Tests Conducted by Detonation Sub-Committee of the Cooperative Fuel Research Committee.

Speaker: G. J. Liddell, Henry L. Doherty & Co.

Afternoon Session: Chairman—H. G. Hoffman, Motor Vehicle Commissioner of New Jersey.

Problems of Industry in Using the Highways.

Speaker: G. E. Clinton, Traffic Manager of Sheffield Farms.

Planning and Administering Highway Systems for Greatest Usefulness.

Speaker: W. J. Sloan, Chief Engineer, N. J. State Highway Dept.

Evening Session: Chairman—

Business Utilization of Air Transport Facilities—Economic Maintenance and Safety Features.

Speaker: E. P. Lott, vice-president, charge of operations, National Air Transport Co.

SATURDAY, NOV. 10

Inspection of Newark Airport and Participation in Dedication Ceremonies of New Administration Building.

DURING THE MEETING

Exhibit of Automotive Equipment.

Diesel and Butane—Propane Burning Units, Streamlined Vehicles, etc.

Tractor & Power Group Plans December Meeting

Taking as its central theme the influence of wear factors in engine design, construction, and operation, the S.A.E. Tractor and Industrial Power Equipment Committee is planning a meeting in Chicago, on Dec. 5 and 6, in cooperation with the meeting of the Power and Machinery Division of the American Society of Agricultural Engineers, which will meet Dec. 3 and 4 in the same place.

In no sense meeting jointly, the divisions of the two societies will cooperate fully in the coordination of their programs, so that tractor and equipment men attending may receive the broadest possible benefit from the conjunction of the two meetings.

Two speakers have already accepted assignments for the S.A.E. sessions; Robert Jardine, chief engineer, Wilcox-Rich Corp., will speak on valves, and K. W. Connor, president, Micromatic Hone Corp., on Cylinder-Bore Characteristics and the Honing Process.

The S.A.E. and A.S.A.E. meetings will be held at the Stevens Hotel in Chicago. C. G. Krieger, agricultural engineer, Ethyl Gasoline Corp., is chairman of the S.A.E. Tractor and Industrial Power Equipment Committee.

New-Type Bus Described by Two

● Northwest

The advanced type of motor coach being put into service between Portland, Ore., and Vancouver, B. C., by the North Coast Lines, was the subject of discussion at the first fall meeting of the Northwest Section.

The meeting was held in Seattle, on the evening of Sept. 14, with 33 present. John M. Holmstrom, who is the chairman for this year, presided.

For the first meeting of the year, with the program arranged at short notice, it proved a successful affair. Murray Aitken of the Kenworth Motor Truck Corp., discussed "The Modern Motor Coach," with the particular coach analyzed being the newest North Coast job. Walter Stark, of the Hall-Scott Motors Co., read a paper on "Horizontal Engines," a Model 135 Hall-Scott, 130 hp. at 2800 r.p.m. type powering this bus.

After the meeting, which ended at 10.30 p. m., nearly all present went for a ride in the motor coach that had been discussed.

The three main features of the bus are: horizontal engine, chassis elimination and electro-pneumatic gear shift.

Mr. Aitken stated that the all-steel bodied coach, chassisless, and with many minor features in construction embodied, weighs about 18,000 lb., or 2000 less than a similar capacity bus of the older, conventional type. The weight saving is largely due to chassis elimination. Height of the bus is 16 to 18 in. lower, and this is a factor in cutting down air resistance.

The "pancake" type Hall-Scott engine is slung from cross-members amidship of the bus, just ahead of the rear axle and just below the upper deck. The driver is seated well forward.

Most radical new feature is the Swift-Newell magnetic gear shift. Mr. Aitken described this as follows:

"To the Mann constant-mesh transmission, for shifting purposes, is applied the electro-pneumatic gear shift.

"Because of the distance of the transmission from the driver, some sort of air shift was found necessary. Gear shifting, in brief, is accomplished by electrical switches, controlling magnetic air valves which in turn introduce air to the proper air cylinders, to make the shifts. The electrical switches are contained in a small box mounted on the instrument panel. In this box is a small gear shift lever which protrudes about 6 in. above its surface. To the lower part of this lever are connected sliding electrical contacts.

"Thus, by putting the lever in the proper desired slot as in conventional shifting, the proper electrical connections are made for the magnetic valves."

"Murder" on Bill at Golfing Event

● Milwaukee

The Milwaukee Section combined golf with "murder" at a frolic held at the Merrill Hills Golf Course, Waukesha, Wis., the afternoon of Sept. 13. About 30 members and guests took part in the golf matches which were followed by a dinner attended by several more persons. The fun at the meeting was provided by Chairman C. E. Frudden and his "secret service" staff. He claimed to have uncovered a vast murder plot and conspiracy. The location of the "crime" remained somewhat indefinite during the whole meeting, but it turned out that the culprit was named "Old Granddad" and he was released on his own recognizance after standing treat to several of the chief investigators.

Ed King came from Chicago to take the low net score in golf, while J. A. Mahoney won the prize for high gross with a score of 133. Past-chairman P. W. Eells was toastmaster at the dinner where the murder investigation was continued under the direction of "Chief" Strehlow.

"Get-Together" Springs Surprise

● Metropolitan

A Metropolitan Section get-together meeting at Spring Lake, N. J., Sept. 7 to 9 was provided with some unexpected excitement by the fact that two life boats from the burning steamer *Morro Castle* arrived on the beach at Spring Lake early on the morning of the 8th. Several of the 34 members and guests of the Society present turned out to watch the rescue operations which pretty well took precedence over all other activities until late in the day.

The banquet on the evening of the 8th was conducted on a "business as usual" basis however, with Sid G. Harris, Chairman of the section as toastmaster, and H. H. Clegg, assistant director of the division of investigation, Department of Justice, as the principal speaker.

Motion pictures were shown of the summer meeting of the general Society and the "proving ground" tests conducted by the Metropolitan Section at its May meeting. Capt. Walter C. Thee, Q. M. Corps, U. S. Army, Fort Hancock, N. J., was chairman of the entertainment committee, which had the largest responsibility in arranging the meeting. Several Members commented with pleasure on the music of Johnnie Johnson's orchestra which played for dancing following the banquet and motion pictures. The three days of the meeting, which was held at the Monmouth Hotel, were chiefly devoted to running off a varied program of sports.

New England Sponsors Electrical Course

Again this year the New England Section of the Society is cooperating with the University Extension Division of the Massachusetts Department of Education in holding an evening class in "Advanced Practice in Automotive Electricity." The course, which is intended primarily for men in the automobile industry, will be given at Massachusetts Institute of Technology on eight Thursdays at 7:30 p. m., beginning Nov. 22, by John J. Gildee, head of the state's department of automotive engineering for the past 13 years.

S.A.E. Nominees for 1935

FOLLOWING are the names of those who have been nominated as officers and members of the Council for 1935:

President ..... **William B. Stout**
President, Stout Engineering Laboratories, Inc.

Treasurer ..... **David Beecroft**
Manager, New York Office, Bendix Aviation Corp.

Councilors

Term of 1935-1936

F. C. Horner
Assistant to Vice-President, General Motors Corp.

H. T. Woolson
Chief Engineer, Chrysler Corp.

Austin M. Wolf
Consulting Engineer, New York.

Members of the 1935 Council will include also J. M. Crawford, Chief Engineer, Chevrolet Motor Co.; J. B. Fisher, Chief Engineer, Waukesha Motor Co.; J. F. Winchester, Coordinator and Supervisor of Motor Equipment, Standard Oil Co. of N. J.; who were elected at the beginning of 1934 for a two-year term; and as Past Presidents, H. C. Dickinson and D. G. Roos.

Vice-Presidents

Aircraft ..... **C. H. Chatfield**
Assistant Director of Research, United Aircraft & Transport Corp. of Conn.

Aircraft-Engine ..... **Philip B. Taylor**
Chief Engineer, Wright Aeronautical Corp.

Diesel-Engine ..... **C. L. Cummins**
President, Cummins Engine Co.

Fuels and Lubricants **D. P. Barnard**
Assistant Director of Research, Standard Oil Co. (Ind.)

Passenger-Car ..... **L. P. Kalb**
Chief Engineer, Continental Motors Corp.

Passenger-Car-Body .. **C. O. Richards**
Body Engineer, Cadillac Motor Car Corp.

Production ..... **V. P. Rumely**
Hudson Motor Car Co.

Transportation and Maintenance ..... **T. C. Smith**
Engineer, Motor Vehicle and Construction Apparatus, American Telephone & Telegraph Co.

Truck, Bus and Railcar ..... **C. O. Guernsey**
Chief Engineer, J. G. Brill Co.

Endeavour's Skipper to Be S.A.E. Guest

Wide World

A tense moment aboard Endeavour with T. O. M. Sopwith at the wheel and Mrs. Sopwith timing.

As the JOURNAL went to press, announcement was received at S.A.E. headquarters that T. O. M. Sopwith, skipper of "Endeavour," eminent British challenger for the America's cup, had accepted an invitation to be the guest of honor at a luncheon sponsored by the Metropolitan Section in New York on Friday, Sept. 28. The luncheon was to be held in the West ballroom of the Commodore Hotel, and to it were invited, in addition to S.A.E. members, members of the American Power Boat Association, the Society of Naval Architects and Marine Engineers, etc.

Apart from being in the public eye as designer and builder of a racing yacht of great interest, Mr. Sopwith has many things in common with the automotive enginering fraternity, as the following brief biography will show.

Thomas Octave Murdoch Sopwith, born in London in 1888, was one of the "early birds" in aviation. He is the possessor of the Royal Aero Club's pilot certificate No. 31, received in 1910. Within four days of receiving his license he established a British record for an endurance flight of 108 air miles in 3 hr., 12 min.

On Dec. 18, 1910, he won the Baron de Forrest prize of £4000 for the longest flight from England to the Continent, reaching Tirlemont in Belgium, a distance of 161 miles in 3 hr.

During 1911 he won several flying competitions in America and in 1912 returned to England, where he founded the Sopwith Aviation Co., and operated a flying school.

From the Sopwith Company came the "Pup", "Camel", and "Snipe" models, famous in early aviation history. One of them in 1912 won the first aerial derby, and in 1913 established a British altitude record.

From 1925 to 27 Mr. Sopwith was chairman of the Society of British Aircraft Constructors. Since 1921 he has been joint managing director of the H. G. Hawker Engineering Co., Ltd.

Revisions in Section Officers List

Since publication in the September issue of the JOURNAL of the names of Section officers for 1935, a number of changes have been reported to headquarters. The status of such changes at the time the JOURNAL went to press was as follows:

● Baltimore

Charles Froesch, sales and service engineer, General Aviation Mfg. Corp., has resigned as chairman of the Baltimore Section and has been succeeded by the vice-chairman, Major James R. Hill, assistant commandant of the Q.M.C. Motor-Transport School at Camp Holabird. A vacancy exists in the vice-chairmanship at present.

● New England

R. R. Whittingham, because of removal to New York, has resigned as secretary of the New England Section. W. M. Clark, superintendent of transportation equipment, S. S. Pierce Co., Boston, is acting as secretary until the appointment of a successor to Mr. Whittingham.

● Washington

Major Stephen G. Henry, U. S. Army Tank Corps, has been transferred to Porto Rico, and has resigned as secretary of the Washington Section. No successor has been appointed.

● St. Louis

G. R. Gwynne has resigned as secretary of the St. Louis Section. Charles A. Marien is acting as secretary until a successor is appointed.

Two Constitutional Amendments Adopted

To the Members:

The amendments to the Constitution of the Society, set forth in the letter ballot dated July 19, 1934, relating to paragraphs C20 and C21, have been adopted.

The report of the tellers follows:

Total number of votes cast		877
Ballots defective		3
	Ballots favoring adoption of proposed amendments	Ballots not favoring adoption of proposed amendments
C20	668	206
C21	655	219

(Signed) Herbert Chase
Joseph A. Anglada
O. P. Liebreich

Paragraphs as Amended

The paragraphs of the Constitution now in effect as amended read as follows:

C20—The Council may, at its discretion, by a three-fourths ballot vote of its duly elected members, change any then existing schedule of initiation fees for membership in the various grades. No increase in fees shall be effective until after two months' notice by publication in the JOURNAL of the Society, or by letter to the membership.

C21—The Council may, at its discretion, by a three-fourths ballot vote of its duly elected members, change any then existing schedule of dues for membership in the various grades, and may provide for the payment of dues for periods of three, six or twelve months. No increase in dues shall be effective unless announced by publication in the JOURNAL of the Society, or by letter to the membership, at least one month prior to the beginning of the period in which the increase is to be first effective.

In the presentation of these amendments it was explained that whereas no changes in dues and initiation fees are contemplated, yet it was felt that the amendments would facilitate making adjustments that might be necessitated by major fluctuations in the value of the currency.

SOCIETY OF AUTOMOTIVE ENGINEERS, Inc.
John A. C. Warner
Secretary

What Members Are Doing

A. J. Brandt been named assistant general manager of the Hupp Motor Car Corp., Detroit. Retained in an advisory capacity by a succession of prominent automobile manufacturers, Mr. Brandt has been president and general manager of A. J. Brandt, Inc., in Detroit.

Sid. G. Harris is eastern branch manager in Brooklyn, N. Y., for the Burton Auto Spring Corp., of Chicago. Mr. Harris, who was formerly connected with the Brandt-Warner Mfg. Co., York, Pa., is chairman of the Metropolitan Section of the Society.

E. D. Herrick's current address is care of Auburn Automobile Co., Auburn, Ind.

James M. Shoemaker is project engineer on power-plant installations for the Chance-Vought Corp., East Hartford, Conn.

Harry E. Ritter retired in August as president and treasurer of the Mid-West Glass Co., Cincinnati. Mr. Ritter writes that his reason for retiring was poor health, and that when he is fully recovered he may again become active.

Gustof L. Hanson has joined the Illinois Tool Works, Chicago, as engineer in charge of broach activities. He was formerly superintendent of the National Broach & Machine Co., Detroit.

Capt. George T. Rolan, Inf. Res., is on active service with the Civilian Conservation Corps at Camp Higgins Lake, Roscommon, Mich. He is executive officer of the 672nd Co., C.C.C.

Burnham E. Field, metallurgical engineer in the research laboratories of the Union Carbide & Carbon Corp., has been transferred to the Niagara Falls Division of the corporation.

Dr. George M. Maverick, manager of the research laboratories of the Standard Oil Development Co. at Linden, N. J., has been transferred to the head office of the company at 26 Broadway, New York City.

Howard S. Manwaring, formerly engineer with the Continental Motors Corp., has joined the International Harvester Co. as research engineer, at Fort Wayne, Ind.

G. Clinton Patrick, formerly with the Rockne Motors Corp., Detroit, has joined the Wagner Electric Corp., St. Louis, Mo.

LeRoy F. LeGros, formerly draftsman with the Chrysler Corp., has been appointed engineer by the Cadillac Motor Car Co., Detroit.

Samuel P. Marley, automotive engineer with the Socony-Vacuum Corp., has been transferred from their research department in Paulsboro, N. J., to their general laboratories at Greenpoint, Brooklyn, N. Y.

Nat Mallouf, formerly president of the Mallouf Haulage & Maintenance Corp., New York City, has been named assistant to the president of the Overman Cushion Tire Co., Inc., New York City. His work will be concerned with the manufacturing and sales activities of the company.

Bjarne Thulin has returned to the Diesel engineering department of the International Harvester Co., Chicago, after a brief period with Purdue University. Mr. Thulin's position is experimental engineer.

John A. White, a past-chairman of the Baltimore Section, has resigned as branch manager in Baltimore for the Mack-International Motor Truck Corp., and is devoting himself to consulting work on operation and maintenance for fleet owners.

Edward W. Dart has joined the Kreider-Reisner Aircraft Co., Inc., Hagerstown, Md., as aircraft engineer.

Peter S. Steenstrup, formerly zone manager at Oakland, Calif., for the Buick-Olds-Pontiac Sales Co., has been named Pacific Coast manager for the Buick Motor Car Co.

Burton W. Wetherbee has been named production chemical engineer of the Russell Mfg. Co., Middletown, Conn.

D. F. Myers is now technical advisor for Cathay Motors, Ltd., Shanghai, China. For the past several years he has been chief engineer of the automotive section of the trench-mortar works at Mukden, Liao Ning, China.

W. H. Beal has been named vice-president of Cord Corp., in charge of manufacturing operations. He will actively assist L. H. Manning, recently elected president of Cord Corp. Mr. Beal has moved forward steadily in the Cord organization for many years, advancing through the sales division to the presidency of Lycoming Motors and then to president of Auburn Automobile Co., the position from which he has moved to his new duties.

W. H. Beal

Morgan J. Hammers has been elected president and general manager of Electrol, Inc., manufacturers of oil burners and air-conditioning equipment. Mr. Hammers is president of the American Oil Burner Association and chairman of the Code Authority for the industry. With his new connection he will move his headquarters to 441 Lexington Ave., New York City.

Howard K. Gandelot, former experimental engineer for the Stewart-Warner Corp., Chicago, has joined the Detroit office of N. W. Ayer & Son, advertising agents.

James M. Shoemaker has joined the Chance Vought Corp. at Hartford, Conn. He was formerly assistant aeronautical engineer at Langley Field, Va., for the National Advisory Committee for Aeronautics.

W. A. Maynard

Walter A. Maynard has joined the Cummins Engine Co., Columbus, Ind., as factory representative. For the past 11 years Mr. Maynard has been with the White Co., most recently as sales manager in the Chicago district and sales promotion manager at the factory.

Warren F. Faragher has joined the Houdry Process Corp., Marcus Hook, Pa. He was formerly research engineer with the Vacuum Oil Co., Paulsboro, N. J.

Alfred M. MacLaren, formerly sales engineer with the SKF Industries of Calif., Inc., has joined Bearing Sales & Service, Inc., Seattle, Wash., in a similar capacity.

Robert W. A. Brewer spent most of September in Europe. He has been working for the last three years on the development of a two-stroke cycle engine.

Fred C. Holz is field service development manager for the Gomery-Schwartz Motor Car Co., Philadelphia. He was formerly a zone manager for the same company.

Charles W. Noller is with the Railway Express Agency as a designer on truck and bus problems.

Peter Menz is employed as a checker by the Wright Aeronautical Corp., Paterson, N. J.

A. M. Oltmanns is in the industrial sales division of the Socony-Vacuum Oil Co., New York City.

Carleton E. Stryker is chief engineer of the Curtiss-Wright Technical Institute of Aeronautics, at the Grand Central Air Terminal, Glendale, Calif. Previous to this connection Mr. Stryker was designer and engineer for the Airplane Development Corp., at Glendale, after

a term with the Northrup Corp. in the same capacity at Inglewood, Calif.

R. W. L. Boname has been named project engineer in the research department of M.

R. W. L. Boname

Louis Breguet, Paris, manufacturer of aircraft. Before returning to France Monsieur Boname was a draftsman with the B/J Aircraft Corp., Baltimore, Md.

O. B. Zimmerman is the author of a paper on "A Technical Analysis of the Agricultural Implement Type of Spoked Wheels," published in the August issue of *Agricultural Engineering*, the journal of the American Society of Agricultural Engineers. The paper, which was presented at the last annual meeting of the A.S.A.E., represents more than 1000 hr. of work, and is of particular interest, in addition to its contents, as being the first of a series of papers on the wheel and other machine elements of fundamental importance to the whole profession. In publishing the paper the A.S.A.E. preceded it with a full-page biographical note on Mr. Zimmerman's career as an "athlete, soldier, scientist and engineer."

Meetings Calendar

S.A.E. Annual Dinner

Commodore Hotel, New York, Jan. 7, 1935.

S.A.E. Annual Meeting

Book-Cadillac Hotel, Detroit, Mich., Jan. 14-18, 1935.

Baltimore—Oct. 4

Canadian—Oct. 17

Royal York Hotel, Toronto; dinner 6:30 P.M. Automotive Engineering Trends—D. G. Roos, chief engineer, Studebaker Corp.; president, S.A.E. Talk by John A. C. Warner, general manager, S.A.E.

Chicago—Oct. 2

Recent Developments in Diesel Engines.

Cleveland—Oct. 15

Cleveland Club; speaker—S. W. Sparrow, research engineer, Studebaker Corp.

Dayton—Oct. 13

Plant inspection trip to Hamilton, Ohio.

Detroit—Oct. 10-11

Book-Cadillac Hotel.

Participation in the Production Meeting and dinner of the Society.

Indiana—Oct. 11

The Athenaeum, Indianapolis; dinner 6:30 P.M.

Automotive Developments of the Army—Major L. H. Campbell, Jr.

Kansas City—Oct. 5

Steuben Club, 35th & Forest Ave., Kansas City, Mo.; dinner 6:30 P.M.

Metropolitan—Oct. 8

Roger Smith Hotel, 40 E. 41st St., New York City; dinner 6:30 P.M.

Recent Trends in Passenger-Car Design—Henry M. Crane, General Motors Corp.

Milwaukee—Oct. 3

Avalon Hotel, Waukesha, Wis.; dinner 6:30 P.M.

Speaker: J. B. Fisher, chief engineer, Waukesha Motor Co.

New England—Oct. 9

Walker Memorial, Massachusetts Institute of Technology, Cambridge, Mass.; dinner 6:30 P.M.

Traffic Surveys, Highway Developments, Progress for Safe Driving, etc.—by a speaker from the Massachusetts Highway Department.

Vacuum Operating of Spark Control—Frank Johnson.

Northern California—Oct. 9

Athens Athletic Club, Oakland; dinner 6:30 P.M. Ten-minute discourses on transportation: Air Transportation—J. F. Long, J. F. Long Co.; Railcar Operation—Mr. Wagner, Southern Pacific Co.; Bus Transportation—L. R. Scholl, service manager, Fageol Truck & Coach Co.; Bay Bridge Traffic Survey—Clinton Veal, Southern Pacific Co.; New Developments in Horizontal Engines—Mr. Patch, Hall Scott Motors; Bus Tires—J. Hunsaker, Gurley Lord Tire Co.

Northwest—Oct. 12

Engineers Club, Seattle, Wash.; dinner 6:30 P.M. Vibration and Balancing—Prof. A. M. Winslow, Engineering School, University of Washington, and H. O. Hanawalt, Engineering Specialties Co.

Philadelphia—Oct. 10

Philadelphia Automobile Trade Association, Inquirer Bldg.; dinner 6:30 P.M.

Oil and Fuel Economies for the Operator—J. C. Geniesse, research engineer, Atlantic Refining Co.

Pittsburgh—Oct. 2

Hotel Schenley; dinner 6:30 P.M. Improved Types of Leaf Spring Suspensions—Karl Probst, chief engineer, Logan Gear Co., and John H. Shoemaker, Code Commissioner of the Leaf Spring Institute.

Southern California—Oct. 12

Richfield Bldg. Cafeteria, Los Angeles; dinner 6:30 P.M. Rear Engine Mounting—Rolla W. Moore, sales dept., White Co.; Operating Characteristics as Developed by Cross Country Run—F. C. Brock, district service manager, General Motors Corp.

Entertainment will be provided, and a new rear-engine mount coach will be available for inspection.

John W. J. Ackermans is body engineer and designer with the Packard Motor Car Co., Detroit. He was with the Olds Motor Works in a similar capacity.

Robert Motion has been appointed assistant manager of the West India Oil Co., Port of Spain, Trinidad, B. W. I. He was formerly service-station manager for the Standard Oil Co. of New Jersey at Newark.

H. C. M. Stevens has been placed in charge of the activities in France of the General Motors Technical Liaison Office in Europe, with the title of consulting engineer. He was chief engineer of the Sunbeam Motor Car Co., Ltd., Wolverhampton, Eng. In his present connection he will make his headquarters in Paris.

W. H. Klocke has been appointed consulting engineer to the E. W. Bliss Co., Brooklyn, N. Y. For a number of years Mr. Klocke has specialized in automobile-stamping problems.

Lon R. Smith, who has been special representative for the Cummins Engine Co., Columbus, Ind., has rejoined the Hercules Motors Corp., Canton, Ohio, as assistant sales manager.

Henry S. Cocklin has been appointed senior aeronautical engineer in the aviation section of the U. S. Coast Guard, Washington, D. C. Mr. Cocklin was formerly project engineer on flying boats for the General Aviation Mfg. Corp., Baltimore.

Charles Dahlquist has also joined the Hercules Motors Corp. and will represent their Diesel-engine line in the automotive and industrial fields, as part of a program of expansion of Diesel activities by the company. Mr. Dahlquist recently has been with the Continental Motors Corp. as sales engineer.

Earle W. Pughe, mechanical engineer with the Chevrolet Motor Co., has been appointed plant manager of the company's Kansas City, Mo., operations.

Fritz Mitschke has moved from 101 Lafayette St., Hartford, Conn., to 191 W. 88th St., New York City.

Harlow Hyde is assistant secretary of the Indiana Motor Traffic Association, Indianapolis.

Finley L. Walton, who has been directing commercial drop-forging sales for the Union Switch and Signal Co., has joined the Champion Machine & Forging Co., Cleveland. Union Switch has discontinued the department, and its unfinished business is being transferred to Champion Machine, where Mr. Walton will direct the department.

Donald L. Bower is assistant to the executive engineer of the Water Section of the newly created National Resources Board. He has been acting as assistant secretary to the Mississippi Valley Committee of the Public Works Administration.

Deaths Reported

Paul Vitz, superintendent of maintenance, Arrow Carrier Corp., Paterson, N. J., died July 19, as the result of an accident.

J. G. Weiss died March 1, at St. Francis' Hospital, Spring Lake, N. J.

Chemical Hay for Mechanical Horses

By Robert E. Wilson[1] and D. P. Barnard[2]

THIS paper represents an attempt to appraise the general automotive-fuel situation from the viewpoint of its relation to farm-tractor fuels. In particular, the probable effects of demands for fuels other than gasoline upon costs of production and distribution have been tied-in with engine operation and maintenance costs for the purpose of indicating the most logical course for fuel and engine development.

In general, the farm-tractor fuel-problem is a very important one from the viewpoint of the tractor user, due to the fact that, as farm tractors are generally used, fuel is a major cost-item amounting to more than one-third of the total cost of operation of the tractor.

In spite of the fact that many other considerations logically outweigh fuel costs in importance —particularly availability of the equipment and its capacity during the seasons of peak demand— this item continues to be one which is scutinized most carefully by the user and emphasized in tractor sales.

The best solution represents, of course, the selection of not only the fuels which represent the cheapest source of usable energy, but also that which permits the best compromise with manufacturing costs, maintenance, and operating difficulties.

Within the last year or so several new fuels for industrial automotive engines have come into prominence. Liquefied commercial butane gas appears to be one of the most interesting of these at this time. In this analysis butane and a number of other fuels, not at present considered of industrial importance, are considered for the purpose of presenting as complete a picture as possible.

It seems apparent from the work which has been carried out thus far that the use of butane and other non-orthodox fuels in such small units as the farm tractor does not appear to be economically attractive.

WHILE the relative economics of "hay-burners" versus tractors varies considerably from time to time and from place to place—as well as from author to author —there can be no question but that the future market for farm tractors can be substantially widened or narrowed, dependent upon whether the costs of tractor operation are reduced or increased. Since fuel, maintenance and depreciation are the primary elements of expense in the operation of a tractor, it is evident that the degree of future prosperity of both the tractor manufacturers and the fuel suppliers is dependent to a considerable extent upon their ability to cooperate in developing a combination of tractor engine and fuel which will permit lower first costs and decreased expense in operation and maintenance. Such effective cooperation requires, first, that a substantial number of fuel manufacturers make a serious effort to meet reasonable tractor-fuel requirements and thus make generally available a fuel which for this purpose is both cheaper and better than kerosene; and second, that the tractor industry put adequate engineering effort on designing and constructing tractors which will really take full advantage of the possibilities of such better fuel.

The situation is somewhat complicated by the fact that the demand for tractor fuel is so seasonal and, in many parts of the country, so small, that distribution costs will be excessive unless the proposed tractor fuel is also suitable for and used extensively in other fields which can share the distribution expense. As indicated hereinafter, it is believed that this problem can be worked out quite satisfactorily under existing economic conditions.

Economics of Tractors Versus Horses

The comparatively easy victory of the gasoline engine over the horse for passenger transportation did not depend on any close matching of costs between the two methods of transportation. It does so happen that the cost of gasoline is on the average just about equal, in cents per million B.t.u. of energy content, to a balanced horse diet of hay, corn and oats. However, the gasoline engine is able to convert into useful work about twice as large a percentage of the energy content of its fuel as is the horse; hence, automobile-fuel costs are sub-

[This paper was presented at the Tractor and Industrial Power Equipment Meeting, Milwaukee, April 19, 1934.]

[1] Vice-president in charge of research and development, Standard Oil Co. (Indiana).

[2] Assistant director of research, Standard Oil Co. (Indiana).

stantially less per horsepower actually developed, though higher per vehicle mile. However, the outstanding advantage of the gasoline engine over the horse is in the *weight* per horsepower developed, which is around 7.2 lb. per hp. for a typical passenger-automobile engine, in comparison with more than 2500 lb. per average horsepower developed by a farm workhorse over an 8-hr. day. This is the basis of the ability of the gasoline-engined automobile to develop flexibility, acceleration and speeds far beyond those attainable by any horsedrawn or horse-powered vehicle.

When, however, we come to consider the farm-tractor problem, these advantages in weight, speed and acceleration are of relatively minor importance; and the farm tractor can only make its way if its total cost of operation over a reasonable period of years means a demonstrable saving to the farmer in comparison with the use of horses. Certainly, comfort and convenience play little part in the farmer's decision, or the low road-speeds, cast-iron seats, and other annoying features of conventional tractors would long ago have been improved.

It is not the purpose of this paper to go into an exhaustive analysis of the economics of tractors versus horses, but merely to present the results of two recent surveys which will serve as a background for the later discussion of the fuel problem as bearing upon this economic balance. References are also given at the end of this paper to a number of other such surveys. Probably the most recent comprehensive survey of the economics of farm tractors is presented in Bulletin 325 of the University of Illinois Agricultural Experiment Station for December, 1933. Table 1, which includes averages of the costs for the three principal types of farm tractors, was taken from that bulletin.

The figures given in Table 1 were obtained from a study of the costs of 116 machines of which 56 per cent were of the general-purpose type, the average use for all machines being in the order of 350 hr. per year. An independent study made

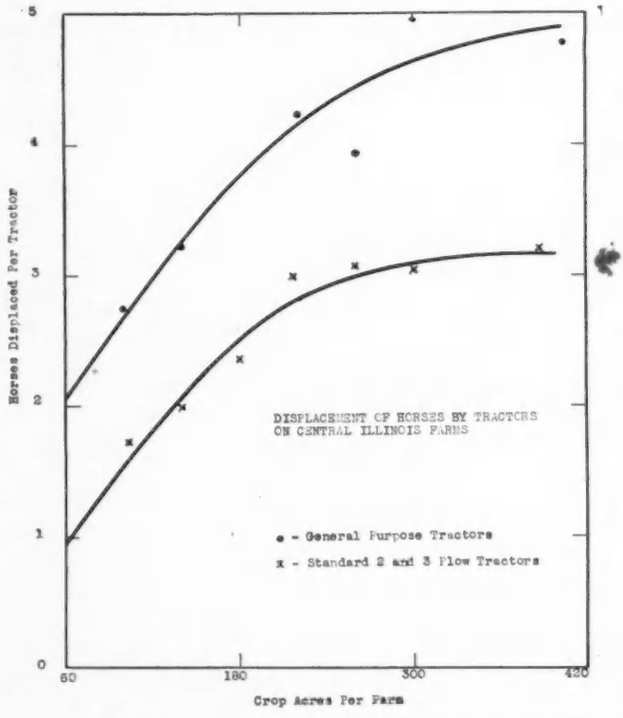


Fig. 1—Displacement of Horses by Tractors on Central Illinois Farms

Table 1.—Average Annual Cost of Tractors versus Horses on an Illinois Farm in 1931

	Tractor		Horse	
	Dollars Per Yr.	Per Cent	Dollars Per Yr.	Per Cent
Operating Costs:				
Fuel and Lubricants	119.63	43.0	39.45	62.3
Repairs, Maintenance, Labor and Miscellaneous	39.52	14.2	.....	....
Fixed Charges:				
Shelter, Depreciation (10-Year Basis), Interest, 6 per cent	118.93	42.8	22.89	37.6
Total	278.08	100.0	62.34	100.0

by the Standard Oil Co. (Indiana) a few years ago, covering the general Midwest farming area, indicated total tractor annual costs averaging $230.62—of which 40 per cent was fuel cost—as against $83.90 for one horse.

Both of these cost comparisons overlook at least two important factors; first, the saving of farm labor, which is very considerable and second, the fact that the tractor can do certain types of work such as the rapid plowing of baked soil in hot weather, which is very difficult, if not impossible to accomplish, with horses. Making some allowance for these factors it would appear that in general a tractor would be justified on a strictly cost basis under conditions where it would replace from 2½ to 3 horses, or even 2 horses in the case of the small general-purpose machine.

As is clearly pointed out in the above-mentioned Illinois survey, the number of horses which can be replaced by one small tractor varies greatly from farm to farm. This number depends upon the size and contour of the farm, the kind of crops raised and various climatic conditions. Of these, the size of the farm appears to be the most important factor and Fig. 1, taken from Bulletin 325, shows the average number of horses replaced by the tractor as a function of the crop-acres per farm. It will be noted from this that if we take 2.75:1 as the ratio of total operating cost—including labor—for tractors versus horses, standard tractors would be justified on farms larger than 200 acres, and general-purpose tractors on farms larger than 100 acres. The most important feature of this curve, however, is to indicate how rapidly the sound economic market for farm tractors varies upward or downward with relatively small changes in tractor operating costs. This emphasizes the vital importance to the two industries concerned of keeping such costs at a minimum if the properly competitive field of the tractor is to be widened and the farmer given his money's worth from his tractor investment.

Without going into further detail as to the exact costs of tractor operation, it is felt that a logical background for a discussion of tractor fuels may be based upon the following assumptions.

(1.) Fuel and lubricant costs equal to 43.0 per cent of total costs
(2.) Maintenance 14.2 per cent of total costs
(3.) Average operation 350 hr. per year
(4.) General-purpose-type tractor 30 belt hp. and 15 drawbar hp.
(5.) Ten-year amortization of equipment
(6.) Engine to operate at 1200 r.p.m.

The point has sometimes been made that part of our crop surpluses are due to the replacement of the hay-, corn-, and oat-consuming horse by the tractor, thus leaving a greater surplus of such farm products to be disposed of elsewhere. Such a contention overlooks two important facts; first, that the horse population has never been a large proportion of the

total farm-animal population—only 9.2 per cent in 1920—and that the decrease in the number of horses in the last 13 years has been more than compensated by the increase in the number of sheep and cattle[3]; and second, that the basic cause of our farm surplus has been the loss of our export trade to other and new grain-producing countries which, by earlier and much more extensive use of tractors and mechanical power in other forms, found it possible to reduce their costs well below ours. If this country is ever to regain its position as an exporter of foodstuffs it is vital that the cost of production be lowered; and that more, rather than less extensive use be made of tractors and mechanical power generally as this is certainly the most promising method in sight for reducing production costs.

Comparative Properties of Possible Tractor Fuels

If a capable research chemist were to be assigned the problem of discovering the ideal liquid fuel for an internal-combustion engine, he undoubtedly would study first the tabulated data on heats of combustion, which indicate the total energy content of any fuel. Of all the known chemical elements, hydrogen shows up with the highest heat of combustion: about 52,000 B.t.u. per lb. Unfortunately, however, hydrogen is a gas which can be liquefied only at extremely low temperatures; in fact, its boiling point at atmospheric pressure is —423 deg. fahr. So hydrogen by itself does not meet the other essential requirements for a convenient liquid fuel.

The chemical element having the next highest heat-energy content is carbon, whose heat of combustion is approximately 14,500 B.t.u. per lb. In this case, however, carbon is a solid which cannot be vaporized to any appreciable extent at temperatures below 6300 deg. fahr., so again this element is scarcely suitable as a convenient fuel for an internal-combustion engine.

As all the other common chemical elements have heats of combustion far below those of hydrogen and carbon, our chemist would readily predict that among the many compounds of carbon and hydrogen, and particularly those rich in hydrogen, would be found the most promising possible engine fuels. A survey of such compounds, known as hydrocarbons, would soon verify this prediction, as there are thousands of hydrocarbons rich in hydrogen which are liquid at ordinary temperature, readily vaporizable to give combustible mixtures with air, and have heats of combustion in the neighborhood of 19,000 to 20,000 B.t.u. per lb. Our chemist could research the entire catalog of more than 400,000 known chemical compounds and not find any compounds having more desirable properties in every respect than such hydrocarbons. If, however, petroleum had never been discovered and the chemist was asked to synthesize such products by any methods available up until recent years, their cost undoubtedly would have been several dollars per gallon; and even the recently developed methods of hydrogenating coal would have resulted in costs in the neighborhood of 40 to 50 cents for such hydrocarbon mixtures unless produced as a by-product in other processes.

It is, therefore, an extremely fortunate fact that this almost ideal liquid fuel can be obtained from petroleum at costs which are of the same order of magnitude—before taxes—as the delivered price of domestic coal per unit of energy content, and much less than the cost of bottled spring water. It is small wonder then that petroleum is the only raw material which need at present be given serious consideration as a source of internal-combustion-engine fuels. The only other appreciable source of hydrocarbons at all suitable for this purpose is coal tar, but the total annual production of motor benzol in the United States in 1932 was less than 0.02 per cent of its gasoline production.

It is true that certain other substances can, under proper conditions, be utilized as automotive fuels. Alcohol is probably the most frequently mentioned of these substances, but the great advantage of the hydrocarbons over other fuels such as alcohol is very clearly illustrated by comparing heating values. Alcohol is really nothing more than the hydrocarbon ethane plus one atom of oxygen. However, the presence of this atom of oxygen really means that the ethane hydrocarbon has already been partly burned or partly combined with oxygen, and its energy content is, therefore, only 12,835 B.t.u. per lb., in comparison with 19,500 B.t.u. per lb. for pure ethane. In addition, as the distillery price for alcohol of adequate purity for blending with gasoline is from five to ten times the refinery price for gasoline, the economic disadvantage of such fuels is clearly apparent.

While the economic superiority of hydrocarbon fuels from petroleum over other possible fuels is very readily demonstrable, the choice of the best petroleum fuel for each different purpose is by no means so simple. Petroleum is a mixture of hydrocarbons boiling all the way from below zero to a theoretical figure of over 1000 deg. fahr. With the exception of the light fixed gases and the very heavy asphaltic residues, practically any of the other petroleum hydrocarbons can be used as fuels in internal-combustion engines of the tractor size; and it is, therefore, desirable to analyze the technical and economic advantages and disadvantages of these possibilities.

The principal commercial products obtainable directly from a mid-continent type of crude without cracking have the characteristics shown in Table 2 with the distillation curves plotted in Fig. 2. Of these products, all except gas oil are fairly uniform in their properties, but the term "gas oil" is applied to almost any heavy distillate from crude petroleum and, unless some other property is specified, it is not at all certain that its other properties will be as indicated. It should further be said that while the above figures for mid-continent distillates apply to about 90 per cent of the products marketed east of the Rocky Mountains, the straight-run products from Pennsylvania crude are higher in A.P.I. gravity and lower in antiknock and heating value than the typical figures given above, whereas products made from most California and certain Gulf Coastal crudes will vary in the opposite direction in these respects.

A study of the individual figures in Table 2 reveals many interesting characteristics and trends. As the boiling point of the cut increases, the A.P.I. gravity of course decreases, the heating value increases, and the knock rating or octane number drops off rapidly, the latter being one of the principal objections to the use of the heavier straight-run distillates in ordinary internal-combustion engines. The drop in the gravity and the increase in the volumetric heating value are both due largely to the fact that the heavier hydrocarbons contain less hydrogen and more carbon than the lighter ones. This relationship between heating value and gravity is so close for the hydrocarbon fuels that it is customary to determine the former from the latter, the relationship determined by the National Bureau of Standards being shown in Fig. 3.

No analysis of the commercial situation relative to petroleum fuels can be significant which does not give very serious and detailed consideration to the economics of cracking and

[3] See Chicago Daily News Almanac, 1934; United States Department of Agriculture Estimates.

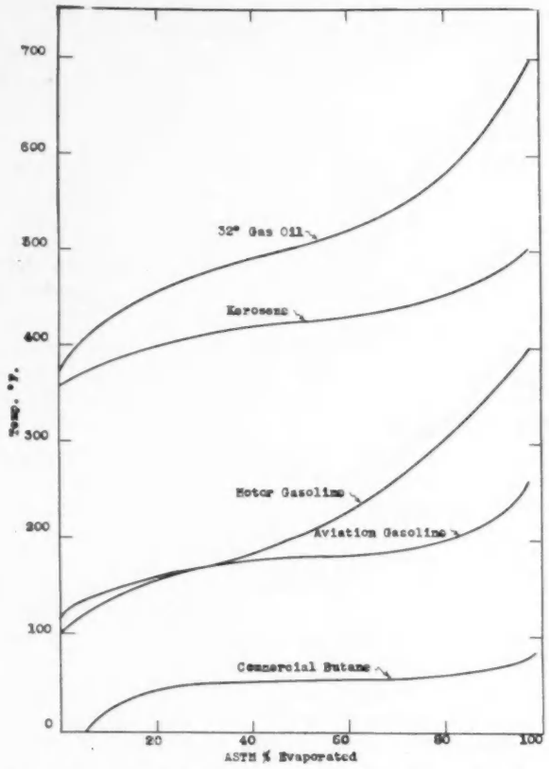


Fig. 2—Typical Distillation Curves of Straight-Run Distillates

the products made thereby. At the present time approximately half of all the gasoline marketed in this country is made from heavier oils by one variation or another of the cracking process. This means that a substantial portion of what could be sold as kerosene, and probably 75 per cent of the gas oil available in most kinds of crude, is subjected to cracking for the production of gasoline. Heavier distillates, including lubricating cuts, and reduced crudes, are also subjected to cracking operations to a very considerable extent and, by and large, it can be said that the true value of any raw distillate heavier than gasoline is today largely determined by its value for use as a cracking stock. In general, the lighter the distillate, the larger the yield of gasoline which can be made by cracking the same, and therefore the greater its commercial value. (Of course, products like kerosene and lubricating oils which are subjected to extensive refining operations are necessarily more costly than the raw distillates referred to above.)

The cracking of these distillates by subjecting them to high temperatures and pressures does not result exclusively in the formation of gasoline. Other products of the primary cracking operation include fixed gases, very light condensable liquids such as propane and butane and related compounds, heavier distillates above the gasoline range, and heavy residues of the fuel-oil and road-oil type. Of these products, the heavier distillates can be, and to a considerable extent are, further cracked for the production of additional quantities of gasoline, but as they give considerably lower gasoline yields than straight-run products of similar boiling range they are to an increasing extent being withdrawn from the cracking system and sold as domestic heating oils, tractor fuels and the like.

Until the last decade, the primary reason for the adoption of the cracking process was to increase the yield of gasoline

Table 2.—Typical Mid-Continent Straight-Run Products

	Butane (Comm'l)	Gasoline Aviation	Gasoline Motor	Kerosene	Gas Oil
Initial Boiling Point, deg. fahr.	....	120	100	350	370
10 Per Cent Evaporated, deg. fahr.	....	145	140	390	430
90 Per Cent Evaporated, deg. fahr.	25	220	350	470	640
Percentage at 392 deg. fahr.	....	....	....	10	2
End Point, deg. fahr.	100	260	390	510	700
Vapor Pressure	65 lb. at 105 deg. F.	7 lb. (Reid)	8 lb. (Reid)	....	....
Flash Point, deg. fahr.	....	....	....	140	140
Octane Number (C. F. R. Engine)	90–100	60–65	50–55	0–10	Below 0
A. P. I. Gravity	113	70	60	42	32
Specific Gravity	0.58	0.70	0.74	0.82	0.87
Lb. per Gal.	4.8	5.84	6.15	6.80	7.21
Heat Content (Net):					
B.t.u. per Lb.	20,500	20,000	19,700	18,600	18,200
B.t.u. per Gal.	97,500	117,000	121,000	126,000	131,500

to meet the rapidly increasing demand. Within recent years the emphasis has been to a much greater extent on the improved antiknock values obtained by this process, which is so important that in most parts of the country straight-run gasoline can today only be sold as a third-grade product at a price disadvantage, while the cracked gasolines are sold as the regular and premium quality, primarily on account of their higher antiknock value. The advantage of cracking is so great that much of the heavier portion of straight-run gasoline, which was formerly sold as such, is now subjected to what amounts to a cracking operation ("reforming") designed primarily not to change the boiling points but to improve the antiknock properties, even at the expense of considerable loss in yield.

The improved antiknock properties of cracked gasoline are due primarily to certain unsaturated and cyclic hydrocarbons

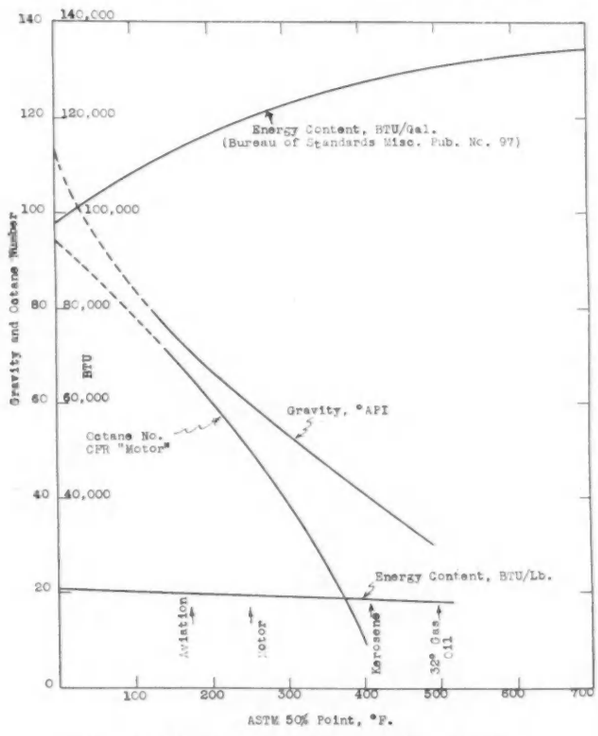


Fig. 3—Mid-Continent Straight-Run Products

formed in cracking, which are not present in ordinary crude oils. This difference in the hydrocarbons is not confined to the gasoline but is characteristic of all the products of cracking, which means that for a given boiling point the products of cracking will have a lower A.P.I. gravity, a lower viscosity, a higher heating value, and a distinctly higher antiknock value. Table 3 and Fig. 4 present data on the characteristics of typical products of these cracking reactions. Of course, different types of cracking processes yield products of somewhat varying character, but these data may be considered fairly typical of most commercial cracked products from mid-continent stocks.

It will be noted that the knock rating of a typical all-cracked gasoline is today around 70 octane number in comparison with around 52 for a typical straight-run product of the same boiling range. This is the main reason why cracked gasolines are preferred to straight-run gasolines for use in ordinary automobile engines. The cracked product does require somewhat more careful refining and frequently the use of antioxidants to prevent gum formation in storage.

The propane-butane cut produced by cracking appears to be somewhat higher in antiknock value than pure butane, but both are so high that this knock improvement is probably not important. On the other hand the difference in knocking characteristics of the heavier fuels is quite outstanding; a cracked No. 1 furnace oil, for example, has an octane number of around 40 in comparison with 0-10 for a typical kerosene. In view of its much higher knock rating, plus its higher heating value and its lower intrinsic value as a refinery cracking stock, it seems inevitable that this type of cracked distillate should, and ultimately will, constitute the preferred domestic furnace oil and tractor fuel, almost entirely replacing kerosene and light gas-oils for these purposes. The use of cracked products in this field again necessitates careful refining, and in general it demands that the end points be kept somewhat lower to prevent the deposition of residues on evaporation either in an oil burner or in an engine manifold. However, the cost of adequate refining is in general substantially lower than that required to make a high-quality kerosene suitable for use in wick-burning lamps and stoves. Even quite heavy cracked distillates can be used in certain types of oil burners

Table 3.—Typical Mid-Continent Cracked Products

	Propane-Butane Cut from Cracked Mid-Continent Stocks	Motor Gasoline	No. 1 Furnace Oil	No. 3 Furnace Oil
Initial Boiling Point, deg. fahr.	...	100	325	375
10 Per Cent Evaporated, deg. fahr.	...	140	380	430
90 Per Cent Evaporated, deg. fahr.	25	350	440	635
Maximum	100	390	490	700
Percentage at 392 deg. fahr.	...	...	15	...
Vapor Pressure	125 lb. at 100 deg. fahr.	8 lb. (Reid)	...	...
Flash Point, deg. fahr.	...	...	120	160
Octane Number (C. F. R. Engine)	100+	70	40-50	0-20
A. P. I. Gravity	120	58	37	28
Specific Gravity	0.565	0.75	0.84	0.89
Lbs. per Gal.	4.7	6.2	7.0	7.40
Heat Content (Net):				
B.t.u. per Lb.	20,600	19,700	18,600	18,300
B.t.u. per Gal.	97,000	122,000	130,000	135,000

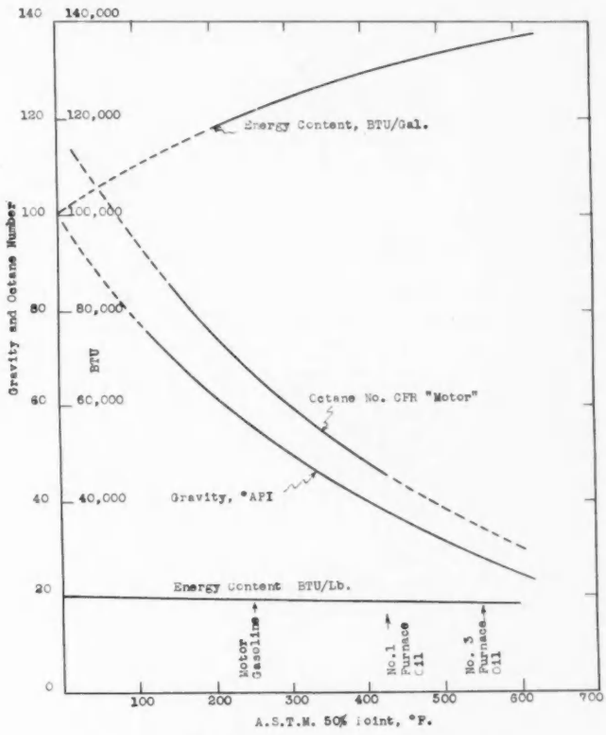


Fig. 4—Mid-Continent Cracked Products

and in properly designed Otto-cycle engines, though in the latter case the dilution problem is more serious and overhauls are likely to be needed somewhat more frequently. The No. 3 cracked fuel described in Table 3 is a still heavier cracked product, which is becoming more widely used as furnace oil in the cities, especially in large installations, and which is available to a limited extent, though not generally, in farming communities.

Economic Factors Involved In Fuel Selection

At the outset of any economic analysis of the tractor fuel situation, the importance of distribution costs must be emphasized. In general, the distribution costs, including freight and all other expenses of marketing, are of the same order of magnitude as refinery prices, and anything which tends to reduce marketing expense is just as important as a similar factor tending to reduce cost of manufacture. The demand for tractor fuel is highly seasonal and the cost of the necessary distributing systems, including tankage, will necessarily be rather high if some special product is demanded. If, however, tractors are designed to utilize fuels which already have a large established market for other purposes, the distribution costs can be shared with a net saving for the tractor user. This is undoubtedly the reason why kerosene has long been considered the principal cheap tractor fuel in spite of its numerous disadvantages. In general, the better a kerosene is for burning in lamps, the higher is its cost, and the lower are its heating value and octane rating and, hence, its suitability for tractor use.

The only other cheap fuel which enjoys wide distribution for other purposes is the previously described No. 1 furnace oil sometimes referred to as "engine distillate." A few years ago this fuel was available only in the larger cities, but within the last three or four years its distribution has grown rapidly and it is now quite generally available throughout the coun-

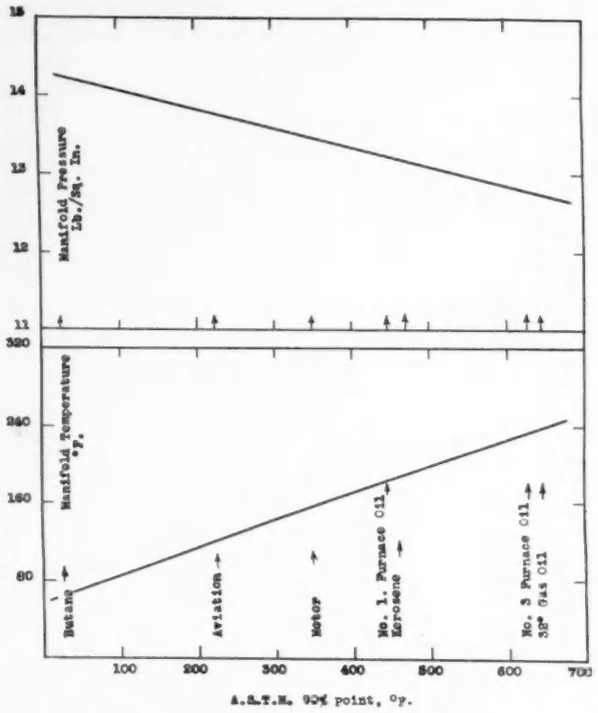


Fig. 5—Calculated Variations in Manifold Pressure and Temperature with the 90-Per Cent Point

try. Unfortunately, by no means all of such distillates are cracked products of high antiknock value because straight-run gas-oils are also marketed for this purpose. As pointed out previously, however, it is not economically sound to sell straight-run products either for furnace oil or tractor fuel because they are less suitable for these purposes than cracked products, and yet they are intrinsically worth more to the refiner, at least in any refinery provided with the usual cracking equipment. The tendency should, therefore, certainly be for cracked fuels to replace more and more completely the straight-run type of product. However, the general distribution of such fuels could be speeded up and operating troubles avoided if the Society or a group of tractor manufacturers would agree to some method of designating tractor fuels which had an octane number in excess of, say, 30 or thereabouts.

The tractor manufacturer also has a very definite part to play in the development of the economics of the tractor-fuel situation. An engine which has a low enough compression pressure and high enough heat on the manifold to handle kerosene or light gas oil satisfactorily is not an engine which will get anything like the available efficiency out of gasoline or a really good tractor fuel. On the other hand, if an engine is designed to utilize efficiently a No. 1 furnace oil of 35 or 40 octane number, it can be used with excellent satisfaction and good efficiency on third-grade gasoline, which probably always will be employed by many users during cold weather or when frequent starting or stopping is involved. To crystallize more definitely the economic factors in fuel selection it is necessary to approximate, first, the efficiency of various engines designed to burn some of the above-mentioned fuels and, second, the delivered price of such fuels.

In evaluating these fuels for tractor service, it was felt that the most reliable appraisal would be obtained if it be considered that, in each case, an engine be designed and built to take the utmost advantage of each fuel within, of course, the practical construction and operation limits imposed by tractor service. It will be noted that the fuels under consideration differ very greatly in the fundamental properties of volatility, heating value, and octane rating as set forth in Tables 2 and 3, with the result that the extremes of engine performance and fuel economy which can be obtained will vary over a wide range. If it be assumed that all of the fuels are to be burned in spark-ignition carburetor-type engines, a fair idea of the relative performance obtainable can be secured by recourse to the well-recognized effects of manifold temperatures, pressures and the like, with, of course, due allowance for compression-ratio variations resulting from different knock ratings.

Of course, the heavier the fuel is, the higher is the manifold temperature required for satisfactory vaporization and distribution; hence, the lower the volumetric efficiency is and the greater is the tendency to knock. In the second place, the greater difficulty of atomization and the greater heat requirements in the manifold necessarily lower the manifold pressure. In the third place, the higher the boiling point is, the lower is the octane number and hence the lower are the compression ratio and the efficiency of an engine which will operate satisfactorily.

The quantitative evaluation of these different factors is, of course, difficult; therefore, it is approached from two angles, first by attempting to predict on a fundamental basis the relative performance which might be anticipated, and then by comparing these "theoretical" curves with actual laboratory and service results.

In working out these evaluations, Fig. 5 represents an approximation of the manifold pressures and manifold temperatures required in practice to vaporize adequately and distribute the fuels of the different 90 per cent points indicated. These values, along with some National Bureau of Standards figures given in Fig. 6 for the relation between

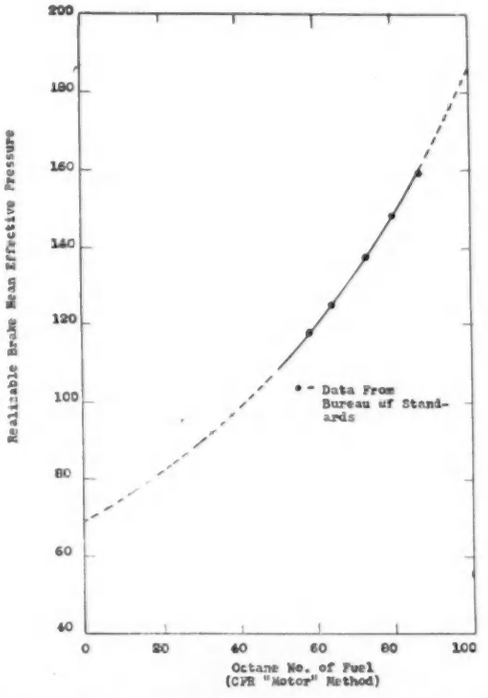


Fig. 6—Attainable Mean Effective Pressures for Different Octane Numbers

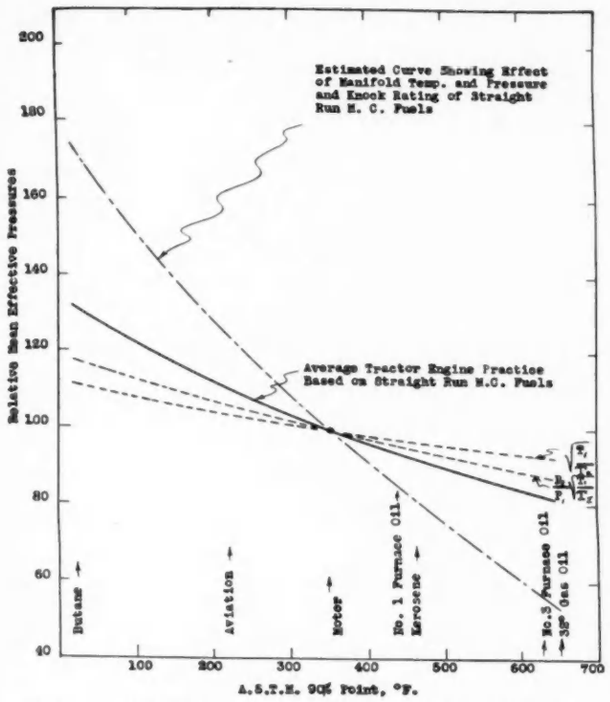

Estimated Curve Showing Effect of Manifold Temp. and Pressure and Knock Rating of Straight Run M. C. Fuels
Average Tractor Engine Practice Based on Straight Run M.C. Fuels
Relative Mean Effective Pressures
A.S.T.M. 90% Point, °F.
Butane
Aviation
Motor
No. 1 Furnace Oil
Kerosene
No.3 Furnace Oil
32° Gas Oil

Fig. 7—Evaluation of the Different Factors Affecting Performance

realizable mean effective pressures and fuel octane-numbers, serve as the basis for the following estimates of engine performance. The evaluation of the different factors affecting performance is then shown in Fig. 7, in which relative mean effective pressures are plotted as a function of the 90-per cent point of the fuel, assuming that the required manifold temperatures and pressures are as indicated in Fig. 5, and that the fuel octane-numbers will be those normally corresponding to mid-continent straight-run products of the indicated volatility. The slope of the most nearly horizontal curve then shows the effect of manifold temperature when the customary assumption is made that engine output varies inversely as the square root of the absolute mixture-temperatures. The difference between this curve and the next steepest one provides for the usual correction of engine output as varying directly with manifold pressure. The curve having the steepest slope shows the "theoretical" combined effect of the three variables: manifold temperature, pressure and the corresponding fuel knock-rating. It will be noted that this curve shows a large variation in specific output for the entire range of fuels under discussion. Actually, it is recognized that mean effective pressures as high as 170 lb. per sq. in. cannot be obtained by recourse to such drastically high compression ratios as would be indicated in Fig. 7, in the case of butane. Therefore, the actual outputs obtainable in tractor practice would not go to such high values if butane were to be used. Further, it is quite possible within the limit of this class of engine practice to avoid the necessity for going to very low compressions with low octane-number fuels by the utilization of such expedients as exhaust-gas dilution, water injection, and the like.

The above-mentioned factors, which tend to reduce the mean effective pressure, naturally increase the necessary weight of the engine per unit power-output. The magnitude of this variation is again difficult to arrive at accurately because of different design standards used in building different types of engines, but a reasonable estimation of the change in necessary engine weight with volumetric efficiency and octane-number of the fuel is shown in Fig. 8, the assumption being that, for this class of service, engine weights will vary inversely as the relative mean effective pressures. While a serious attempt has been made to obtain the best possible figures for the foregoing curves, they can only be semi-quantitative, and the results of any two engine builders will doubtless differ materially. They serve, however, to give a reasonably accurate picture of the fuel situation and of the sacrifice in engine outputs which must be expected if the heavier fuels are to be utilized.

Inasmuch as in tractor operation fuel costs are more important than ordinary differences in engine output or weight, it becomes necessary to estimate as closely as possible the fuel consumptions which would accompany the variations in performance in the foregoing figures. For the purpose of the present discussion, this has been done on the basis that increased engine output obtained by compression-ratio increases will in general be accompanied by a corresponding decrease in *specific* fuel-consumption. Where such output improvements are obtained through the medium of increased volumetric efficiency, however, it is assumed that actual fuel-consumption will increase with air consumption, but that the improved efficiencies of fuel utilization will be such as to offset such an increase to the extent of 50 per cent of the extra fuel which would otherwise be used. It is recognized that such assumptions will again not agree exactly with the results obtained on any particular engine. It does appear, however, that, in general, they cannot be very far wrong, and the sum-total effect of applying these assumptions does produce estimated fuel consumptions agreeing quite closely with those currently being obtained by the best practices with which the writers are familiar.

When the curves for effects of different factors upon variations of mean effective pressure are interpreted in this manner,

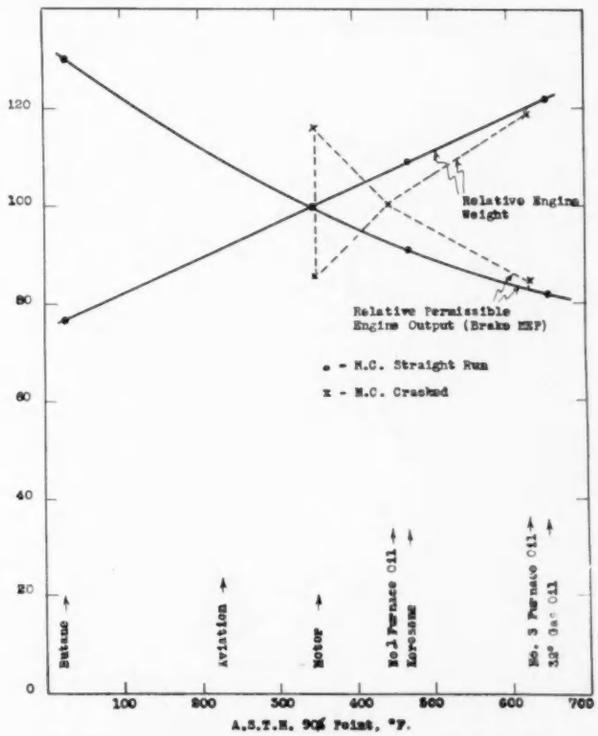


Fig. 8—Estimation of the Change in Necessary Engine Weight with Volumetric Efficiency and Octane Number

the relative fuel-consumption curves as given in Fig. 9 are obtained. Here it will be noted that, as is generally recognized, the heavier fuels cannot be utilized as efficiently on a thermal basis as can the lighter ones. Nevertheless the increased specific gravity and, therefore, energy content per gallon, of the heavier fuels more than offsets the lower utilization efficiency with the result that consumption on a volumetric basis tends to decrease as heavier fuels are used. Again, in preparing Fig. 9, the curves have been drawn upon the basis of straight-run products only. The points located for a cracked No. 1 furnace oil of 40 octane number show very definitely lower fuel consumptions than would be realized with the corresponding straight-run product. The very marked improvement in volume-basis fuel-consumption is, of course, due to the combination of the two factors of greater energy content and the improved utilization efficiency accompanying an octane number of 40 as compared with approximately zero of the straight-run material.

The principal remaining question to be determined is the effect of fuel costs on these comparisons. In many analyses of this type, and particularly in comparing Diesel engines with spark-ignition engines, the mistake has been made of comparing tank-car or even refinery prices of by-product fuels with retail service-station prices, including taxes for gasoline. Relative costs of 15 cents for gasoline, in comparison with 5 cents for gas oil, which appear to be figures commonly used, must have been arrived at in this way. They do not represent any properly comparable prices, but do give rise to misleading conclusions with regard to relative fundamental operating costs of Diesel versus gasoline engines. If the by-product fuels are to be distributed at all extensively, they must carry similar, although somewhat lower, distribution costs compared to gasoline, and if they are to be used in vehicles on the road, they will in most cases be required to carry taxes similar to those paid on gasoline. On the other hand, if used in farm tractors, in almost all the states using tractors extensively, gasoline is tax exempt as, of course, are the heavier fuels.

It is difficult to give any accurate picture representing average prices over a considerable period of time; but, for comparative purposes, the actual present prices of various petroleum products delivered on the farm by tank wagon—exclusive of tax—have been employed. Table 4 accordingly gives the prices in effect on March 26, 1934, first, in tank cars at Oklahoma refining points, and second, a delivered market price at Cherokee, Iowa, a typical farm community so far as freight and distribution costs are concerned. The difference between these two figures indicates the freight and distribution costs, plus marketing profit—if any—for the various fuels on the date in question. It appears that the tank-car prices prevailing at this time reflect fairly well the approximate relative costs of manufacturing the various products from $1 mid-continent crude in a fairly efficient refinery, but include no appreciable amount of profit. The fundamental reason for the lower price of the heavier fuels is, of course, the decreasing amount of gasoline which can be made from these heavier fuels, particularly the cracked distillates. The difference in freight and distribution costs, and the like, between the lighter and heavier fuels is somewhat difficult to explain quantitatively. There is, of course, some difference in freight, and in the handling of gasoline there is an evaporation-loss problem and the necessity for seasonal adjustment of volatility which tends to increase gasoline distributing costs somewhat. Also they possibly reflect in part the rather high cost of service-station distribution. It would appear that the difference in distribution costs between gasoline and the heavier fuels is somewhat larger than sound economics would justify and that in time this differential might tend to decrease somewhat with the result that the heavier fuels would enjoy slightly less price advantages over gasoline than are indicated in Table 4; however, the figures in this table are considered representative and have been used in the ensuing calculations.

Table 4.—Representative Tank Wagon and Refinery Prices of Actual and Assumed Tractor Fuels

	Tank-Wagon Price Exclusive of Tax at Cherokee, Iowa, March 26, 1934. Cents per Gal.	Refinery Price Job Group 3, March 26, 1934. Cents per Gal.	Freight and Delivery Charges Cents per Gal.
Butane in Cylinders (estimated)	14.6	....	10.600
Aviation Gasoline[1]	13.0	5.375	7.625
70 Octane No. Motor (Red Crown)	12.5	4.875	7.625
50 Octane No. Motor	11.0	4.125	6.875
Kerosene	10.1	3.375	6.725
No. 1 Furnace Oil	7.6	3.00	4.60
No. 3 Furnace Oil	6.6	2.50	4.10
32 Deg. Gas Oil	7.0	2.90	4.10

[1] Aviation gasoline is actually sold at a somewhat higher price because of limited distribution.

Combining these fuel-cost figures with the values previously given in Fig. 9 for relative volumetric fuel-consumptions gives the results shown in Fig. 10 for the relative fuel cost, ex tax, for various possible tractor fuels. These figures indicate that the heavier fuels have a very substantial advantage in the fuel cost alone, and we must next determine to what extent these fuel savings are offset by other factors. As pointed out previously, the heavier the fuel is, the heavier the engine will be, the greater the tendency toward oil dilution, the greater the lubrication and maintenance costs and the less easily the engine will start. It also should be remembered that engines operating on the heavier fuels must in general be started on gasoline, which may appreciably reduce the savings to be made on the assumption of 100-per cent operation on the

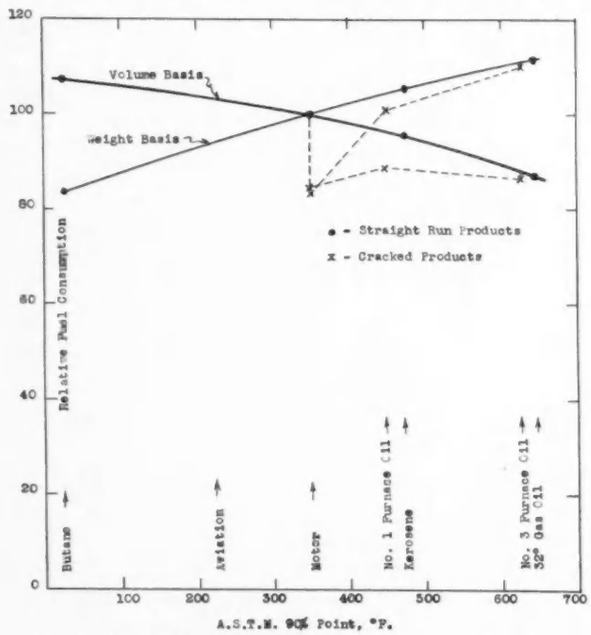


Fig. 9—Curves for Fuel Consumption versus 90-Per Cent Point

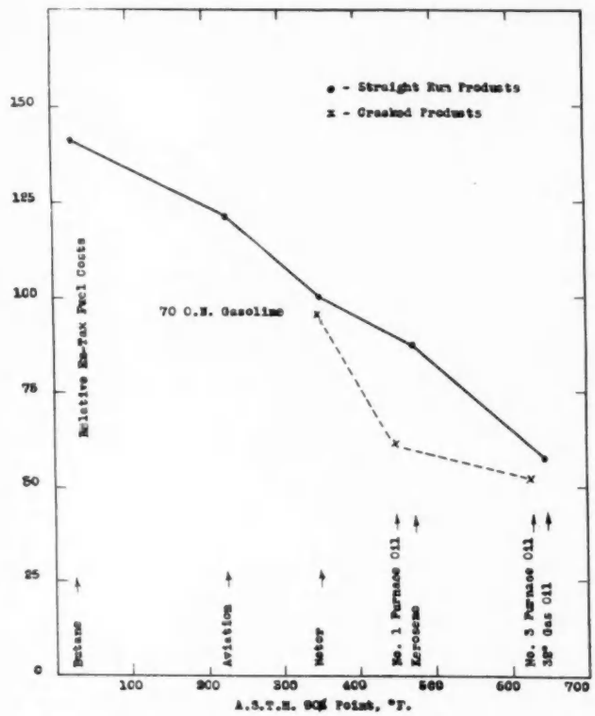


Fig. 10—Curves for Fuel Cost versus 90-Per Cent Point

heavier fuel. While it is difficult to estimate all factors quantitatively, the authors have made the best approximations they could in arriving at the final estimates for total tractor operating costs on the various fuels in question, which are shown in Fig. 11. This comparison, of course, greatly reduces the percentage advantage of heavier fuels particularly on present-day prices, but it will be noted that the balance still appears to be in favor of the heavier fuels.

It must be pointed out that certain intangible factors are not taken into account in the above-mentioned comparison; for example, the effect of engine weight upon the total tractor weight and performance has not been taken into account, nor has the greater convenience of using a single fuel like gasoline for starting and operating been given any quantitative value. Also, the increased maintenance costs always attendant upon the using of higher manifold temperatures and heavier fuels may result in the tractor being unavailable at some time when it is particularly needed, so that the advantage shown for the heavier fuels is still partly offset by intangible convenience factors.

One outstanding conclusion to be drawn from the cost curves is the great advantage of a cracked furnace oil over kerosene as tractor fuel. The reasons for this advantage, namely, lower cost and higher octane number, have been discussed previously and it would appear from this that there is no excuse for designing tractor engines to operate on kerosene where a good-quality cracked furnace-oil is available. Of course, to obtain these higher efficiencies, it is necessary to design the tractor engine to take advantage of higher knock ratings, which means that it will not be suitable for operation on kerosene unless means are provided to minimize knocking. It could, however, be operated on third-grade gasoline of a slightly higher octane number with excellent satisfaction. Past experience indicates that, in many cases, particularly in winter, the increased flexibility and decreased maintenance resulting from the use of gasoline results in the farmer using it in many cases where it is not justified by a bare cost analysis, especially in the low-compression engines designed for kerosene, where it is impossible to take advantage of the real possibilities of gasoline as a fuel. It appears that most farmers are willing to pay 15 to 20 per cent more for gasoline than for heavier fuel an account of these convenience and maintenance factors.

There is another occasion where the use of gasoline is frequently found advantageous in engines designed for heavier fuels, and that is in cases of plowing hard-packed soil, which demands an increase in the power output of the tractor. This can readily be obtained by the use of gasoline with the correspondingly lower manifold temperatures made possible thereby. Under such circumstances, it may become a definite economy to employ a more expensive and volatile fuel to secure the increased tractor power along with saving in warming-up time, servicing and oil draining, and the like.

It is the opinion of the authors that more attention might be well given to the reduction of the weight of tractors, which would increase their utility on hilly surface or muddy or boggy soil. In the cases where higher tractor weight may be advantageous to improve traction, it can be easily obtained at a low cost by adding a few pieces of pig or scrap iron for the temporary situation where needed.

While the use of relatively high-antiknock gasolines is seldom justified in engines designed for low-antiknock fuels, tractor manufacturers should not overlook the fact that an engine really designed to use a fuel of 70-octane number would give an increase in power output per unit weight of 10 per cent in comparison with cracked No. 1 fuel oil, or about 15 per cent in comparison with kerosene. If proper advantage—including the use of high-speed engines—is taken of these differences in power output and efficiency, a truly light-weight powerful tractor could be designed which would seem to have real value, at least for certain classes of service.

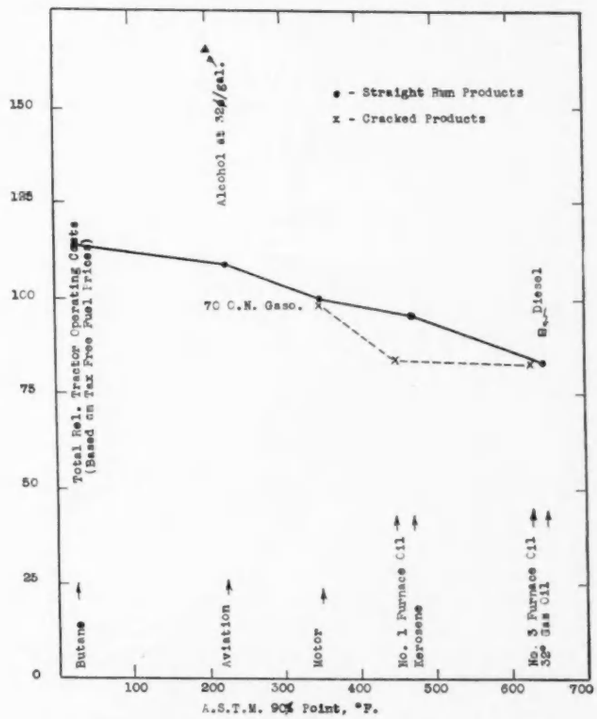


Fig. 11—Curves for Total Operating Costs versus 90-Per Cent Point

Of course, it would have to be clearly stated that such an engine was designed only for the use of substantially 70-octane gasoline.

In Fig. 11, covering total operating cost of different fuels, are given two additional points, one for the use of alcohol and another for the use of a Diesel-engined tractor. In the latter case but little information is available as to engine costs; but, from values quoted in the 1934 statistical issue of *Automotive Industries,* it appears that a Diesel-engined tractor—in size much larger than the one assumed for this comparison—will cost at least 30 per cent more than an equivalent carburetor-engined tractor. This has been confirmed by Jahnke, who has indicated increased *Diesel engine* first costs of 50 to 75 per cent above the equivalent carburetor engine. All injection engines suffer from the high costs of their fuel pumps and the operating difficulties which develop if particles of dust or dirt get into the fuel system. Also, maintenance on Diesel engines is reported to be definitely higher than for gasoline engines by about 30 per cent. As a result, it appears that Diesel power is not particularly promising for small tractors, and even in the larger sizes only where much greater use is made of the equipment than the average of 350 hr. per year. While Diesel-powered tractors may in some cases show fair costs in comparison with gasoline or kerosene-powered tractors, it is not believed that they can show any advantage over tractors with engines designed to use a cracked tractor-fuel, unless the average operating time per year is well over 1000 hr. Most Diesel engines for farm use also suffer from the necessity of requiring for best results a straight-run gas-oil rather than a cracked gas-oil, which latter, for reasons indicated earlier in this paper, will tend to be cheaper and more generally available than the straight-run product.

Varying Demand and Future Fuel Prices

The true economic picture of the internal-combustion engine fuel-situation cannot be based entirely on current market prices, as relative demands sometimes serve to obscure the true picture of fundamental fuel-costs. Basically, the present price-structure depends on the fact that gasoline is the principal product for which crude is run, and that gasoline demand exceeds the straight-run product available, thus necessitating cracking. To illustrate the effect of varying demand, commercial butane was recently available at a very low figure on the West Coast because it could not all be used in gasoline. This resulted in a rapidly increasing demand for it for highway-transport work, which necessitated its withdrawal from motor gasoline. Its value immediately becomes that of a very high-octane-number gasoline which could only be sold profitably at a considerable premium. To a greater or less degree, this is true of all petroleum products and, should the demand for any one become so predominant as to require the running of crude for its particular manufacture, its costs will, of course, tend to increase.

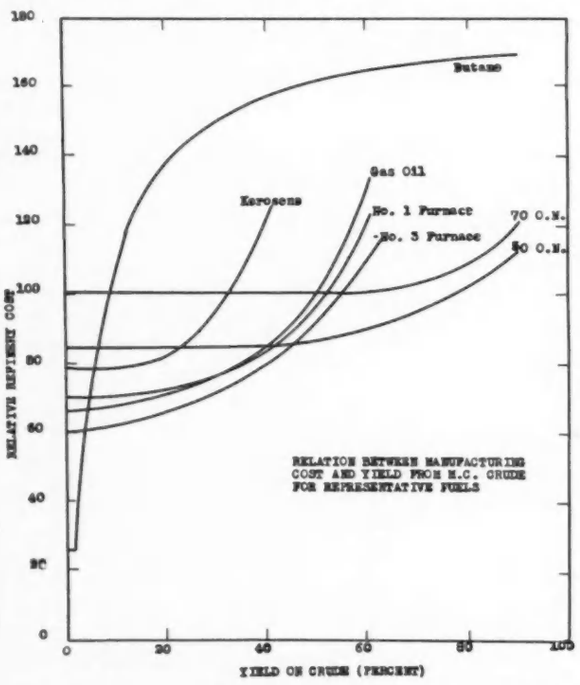


Fig. 12—Curves Showing Relative Refinery Costs Which Probably Would Prevail as the Demands for the Individual Products Increase

This point is illustrated in Fig. 12, which shows roughly the relative refinery costs—based on those of 70-octane-number gasoline—which probably would prevail as the demands for the individual products increase. In preparing this chart, it was, of course, necessary to make many simplifying assumptions, including basing the calculations upon average mid-continent crude. It is believed, however, that the essential trends indicated are substantially correct. As far as *manufacturing costs* are concerned, the chart portrays quite graphically the fallacy in assuming that any engine requiring virgin gas-oil can replace the gasoline engine to any great extent on a fuel-cost basis, because the gas-oil price would increase sharply if more than 30 per cent of the crude were to be used in this way. Even in the case of the furnace oils, if the relative demands for such fuels amounted to but about two-thirds of those now existing for gasoline, it would force their costs up to those now in effect for gasoline. The fact that the efficiencies of utilization for the various fuels vary materially alters the picture slightly but not to an important extent.

It will be noted from the form of the butane curve in Fig. 12 that only a relatively small amount of this material can be made available without an inherent rapid increase in its cost. The exact amount is difficult to predict with accuracy at this time. It appears, however, that the supply of butane now available at by-product costs cannot amount to more than 2 per cent of the gasoline production. This is due to the fact that to produce any greater amount than this would require its withdrawal from gasoline, where, because of its high octane-number, it is a very valuable stock just as long as vapor-pressure restrictions are not exceeded. Coupled with what appears to be an excessively high distribution cost, if butane is to be handled in small quantities at irregular periods such as would be required in connection with its use as a tractor fuel, it does not seem that there is any hope for its economic use in the farm-tractor industry. This does not mean that butane is not a very desirable engine fuel under some circumstances. In fact, it appears that there may be a rather limited number of very important industrial applications for butane, one of the most promising of which may lie in its use as a fuel for railcar and switching-locomotive service. Here the material need only be supplied at a comparatively small number of fueling points and in tank-car quantities. This should tend to reduce greatly the handling and distribution charges, which, when taken into consideration with the fact that butane makes possible some very desirable large-engine design-features, apparently indicates it to be a rather attractive possibility for this class of service.

Effect of Tax Factors on Tractor Fuel Specifications

The foregoing conclusions have been based upon the fundamental economies of costs of manufacture and distribution which, of course, make no allowance for such sales taxes as are now generally imposed on motor gasoline. As fuels for tractor use are generally exempt from State taxes—as they certainly should be—these conclusions should be directly applicable to present practice. To consider the quantitative effect of the taxes which are imposed becomes impractical, due to the great lack of uniformity which exists between different states. The assumption of a fixed tax-rate per gallon upon the various fuels considered, however, would further penalize butane and the lighter fuels and would only serve to emphasize the desirability of using the heavy fuels so long as the present balance of demand is not disturbed and as long as greatly increased engine costs are not incurred. This effect of the road tax in penalizing the low-heat-content fuels, such as butane and alcohol, serves to devaluate them quite appreciably and place a practically unattainable premium on utilization efficiency if they are to compete with the heavier fuels for tractor service.

It appears from the foregoing analysis that, if the basic requirement of lowest possible tractor power-cost is to be met, a fuel heavier than gasoline should usually be employed. Just how much heavier it should be is rather difficult to determine with exactness. It is rather doubtful if the general use of a fuel as heavy as No. 3 furnace oil can be justified, as the more frequent overhauls required may interfere with capacity of labor which is dependent upon the tractor. It is quite impossible to evaluate definitely this important point, but some work has been done which indicates that the greater flexibility of the lighter fuels has a very definite value to the tractor user, probably amounting to at least 10 per cent of the tax-free selling-price of No. 3 fuel as compared with No. 1. Probably, therefore, the most economic fuel to the user of the mechanical horse at this time would correspond quite closely to No. 1 cracked furnace-oil.

To be sure of obtaining a satisfactory product for a properly designed engine a minimum octane-number specification of 30 would appear desirable. To assure adequate volatility and avoid the necessity for excessive manifold heat or long warming-up periods, it would be desirable to specify that at least 10 per cent of such fuel boil below 392 deg. fahr. and at least 90 per cent boil below 475 deg. fahr. Even lower specification points than these would probably be desirable from the standpoint of operation, but it must be kept in mind that furnace oils and tractor fuels should, so far as is possible, be kept in the wholly tax-exempt class which necessitates meeting very definite legal restrictions in most states. In some states, the tendency has been to insist that the 90-per cent point be *above* 475 deg. fahr., but this specification is undesirable both from the standpoint of producing a good tractor fuel which will not deposit gum, give a smoky exhaust, or cause excessive dilution, and also from the standpoint of making a cracked furnace-oil which will not give trouble in certain burners. Further, specifying a high 90-per cent point does not have much to do with the question of whether or not tax evaders can blend the fuel with casinghead or other light gasoline for use in automobile engines. It is, of course, impossible to make a tractor fuel or furnace oil which cannot be mixed with gasoline to some extent; but to minimize these possibilities the two most important things to specify are, first, that the 90-per cent point shall be above, say, 440 or 450 deg. fahr., so that any blend can readily be distinguished from gasoline by the ordinary A.S.T.M. distillation test; and second, to insist that the fuel have a minimum quantity of light ends in the gasoline range so that it will not start in an automobile engine. This is generally done by specifying that the flash point shall be above 100 deg. fahr.—closed cup—and that either the initial boiling point be above 300 deg. fahr. or probably better by specifying that not more than 20 to 25 per cent shall boil below 392 deg. fahr., which marks about the top limit of most commercial gasolines. Such limits on the front-end volatility of a tractor fuel do make it difficult to start a tractor engine on such a fuel, even in hot weather, but the best engineering opinion on this subject is to the effect that even if engines can be started on the tractor fuel they *should not* be, because of excessive dilution and wear which is likely to occur during the warming-up period with any fuel boiling above the gasoline range. For this reason inability to start directly on the heavy fuel is probably a desirable, rather than an undesirable, feature in such fuels.

The Western Petroleum Refiners Association recently adopted the following recommended specifications for tax-free tractor-fuel. A committee of the American Petroleum Institute has also recommended them.

	Deg. Fahr.
Flash Point not below	110
10 per cent not below	347
25 per cent not below	392
95 per cent not below	464
Color	-16 or darker.

These specifications are in general accord with the previously discussed recommendations except that they do not include an octane-number specification.

Probably one of the most effective forms of cooperation between the tractor and automobile industries would consist in making certain that specifications for tax-free fuels do not contain too high a minimum limit on the 90-per cent or the 95-per cent points and to encourage the states to make reasonable restrictions limiting the amount of light ends present in the tractor fuel. The recently adopted Illinois regulations specifying a flash of 100 deg. fahr.—closed cup; a 300-deg. fahr. minimum initial boiling point; not more than 25 per cent off at 392 deg. fahr.; and not more than 90 per cent off at 450 deg. fahr. seem logical and desirable as a solution of this problem. North and South Dakota have similar though slightly less strict specifications.

It is concluded, therefore, that a tractor fuel of the above suggested volatility characteristics and of at least 30 octane number represents the most satisfactory all-round compromise to be reached at this time.

Bibliography on Tractor Costs

Chicago Daily News Almanac, 1934; United States Department of Agriculture Estimates.

United States Department of Agriculture Bulletin No. 997; Cost and Utilization of Power on Farms Where Tractors Are Owned.

United States Department of Agriculture Bulletin No. 1447; Cost of Using Horses, Tractors and Combines on Wheat Farms in Sherman County, Oregon.

United States Department of Agriculture Farmers Bulletin No. 1297; Cost of Using Tractors on Corn-Belt Farms.

United States Department of Agriculture Farmers Bulletin No. 1296; Changes Effected by Tractors on Corn-Belt Farms.

United States Department of Agriculture Farmers Bulletin No. 1295; What Tractors and Horses Do on Corn-Belt Farms.

International Harvester Co. Report, Chicago; Muscles or Motors.

Fin and Cylinder-Baffle Design for Air-Cooled Engines

By Carlton Kemper

Mechanical Engineer,
National Advisory Committee for Aeronautics

THE National Advisory Committee for Aeronautics has undertaken a research to determine the effect of the fundamental factors influencing the cooling of air-cooled radial engines. This paper presents some of the results of an investigation made to obtain information that can be used in the design of fins for the cooling of heated cylindrical surfaces in an airstream.

An equation has been developed for calculating the quantity of heat dissipated by a given fin-design, using experimentally determined heat-transfer coefficients. The agreement between the quantity of heat dissipated, as determined by experiment and calculated by the equation, was found to be sufficiently close to justify its use.

A method has been developed for determining the fin dimensions, utilizing a minimum of material for a variety of conditions of heat transfer, airflow, and metals. The design of cylinder baffles has also been investigated and results are presented that show the improvement in cylinder-temperature distribution and the increase in the average heat-transfer coefficient to be obtained by the use of shell and integral cylinder-baffles.

THE power output of internal-combustion engines is limited by the quantity of waste heat that must be dissipated to the cooling medium. For air-cooled engines at full throttle, the quantity of heat to be dissipated by the fins to the cooling air is equivalent to approximately 45 per cent of the brake horsepower. The problem of the engine designer is to dissipate a definite quantity of heat from a given cylinder at a specified airflow and maximum fin-temperature and, at the same time, to keep the weight of the cylinder to a minimum.

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

[1] See N.A.C.A. Technical Report No. 158, 1932; Mathematical Equations for Heat Conduction in the Fins of Air-Cooled Engines, by D. R. Harper, 3d, and W. B. Brown.

[2] See N.A.C.A. Technical Memorandum No. 725, 1933; Development of Air-Cooled Engines with Blower Cooling, by Kurt Löhner.

Little published information is available on the heat transfer from a finned metal-cylinder to a cooling airstream. Harper and Brown[1] have developed theoretical equations for the heat transmitted through fins when the surface heat-transfer coefficient is known and is constant over the surface of the fins. The heat dissipated from a number of finned cylinders mounted in a free airstream has recently been determined experimentally by Löhner[2].

The National Advisory Committee for Aeronautics is now conducting an investigation on the cooling of finned cylinders which will include the experimental determination of the heat transfer from finned metal-cylinders in an airstream, the effect on the heat transfer of baffles around the cylinder, and the measurement of the air quantity and blower power required to cool satisfactorily a given design of engine cylinder. This paper presents some of the experimental data recently obtained, dealing with the heat transfer from finned cylinders in an airstream.

Test Cylinders

The relative advantages of different fin-shapes and the experimental surface heat-transfer coefficients have been determined from wind-tunnel tests of 18 finned steel-cylinders. The cylinders had an inside diameter of 4.5 in. and a wall thickness of 0.08 in. The fin dimensions investigated included a range of fin pitches from 0.10 to 0.60 in., fin widths from 0.37 to 1.47 in., and average fin thicknesses from 0.04 to 0.27 in. All the fins had tip radii of 0.02 in.

The range of airspeeds used in the investigation of the finned cylinders was from 30 to 150 m.p.h. A smooth cylinder of the same diameter as the finned cylinders was tested at airspeeds from 15 to 215 m.p.h. Fig. 1 shows the principal dimensions of the wind tunnel used in this investigation. As the heat transfer is a function of the product of the speed and density of the air, all airspeed readings were corrected to a common density which corresponds to an atmospheric temperature of 80 deg. fahr. and a barometric pressure of 29.92 in. of mercury.

The cylinders were electrically heated and guard rings were used to eliminate heat flow through the ends of the cylinder and to obtain two-dimensional airflow over the test section. The heat input for the cylinders ranged from 23.4 to 101.3, B.t.u. per sq. in. of outer cylinder-wall area per hr. Temperatures were measured by means of 24 iron-constantan thermocouples spot-welded to the cylinder wall and the fins. The average surface heat-transfer coefficient (q) was calculated by dividing the measured heat input by the product of the total

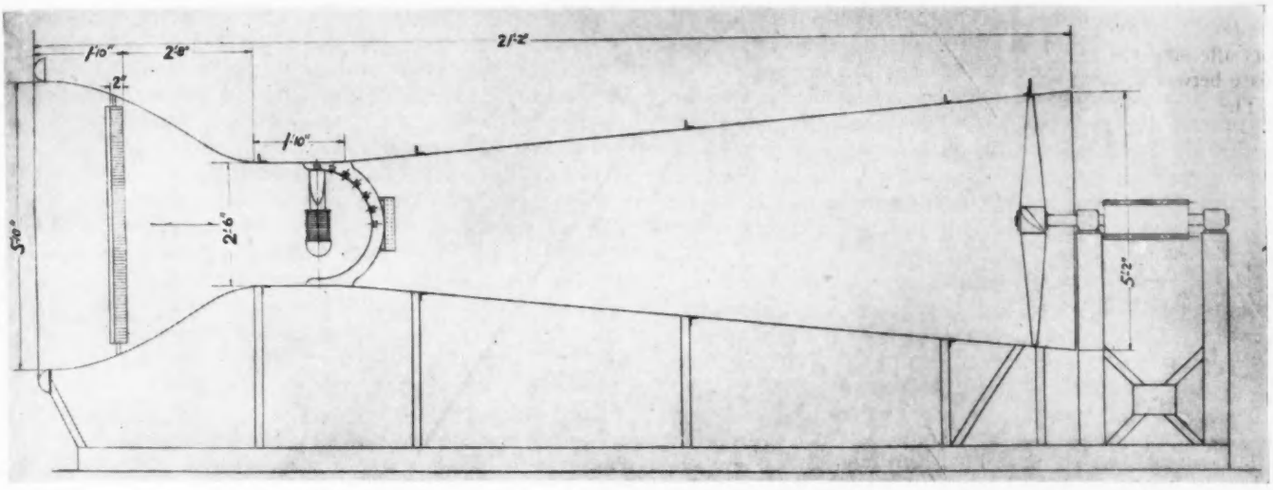


Fig. 1—Wind Tunnel Showing Finned Cylinder Mounted for Testing

cooling surface and the average temperature-difference between the surface and the cooling air. A complete description of the apparatus and methods used in this investigation are given in a recent Committee publication[3].

Airflow

An indication of the probable airflow around the finned cylinder was obtained during previous experiments[4] by coating the adjacent cylinder and guard-ring fins with a mixture of lamp black and kerosene and placing them in the airstream. Fig. 2 shows the pattern obtained at an airspeed of 76 m.p.h. The break-away of the flow from the cylinder occurred at an angle of approximately 115 deg. from the front of the cylinder. The flow pattern indicates dead-air regions at the front of the cylinder and to the rear of the break-away position.

The curves of heat-transfer coefficients q, plotted for positions around the cylinder as in Fig. 3, also prove the existence of these dead-air regions. It may be noted that, for the smooth cylinder with the axis parallel to the flow, the values of q at the rear of the cylinder were found to be higher than those at the front. Similar results were obtained when the tests were repeated with the cylinder rotated through 180 deg. to eliminate any inequalities in the heating.

Surface Heat-Transfer Coefficients

The surface heat-transfer coefficient of finned cylinders depends upon the velocity distribution between the fins and is a function of the free airstream velocity and of the fin dimensions that materially influence the airflow. Biermann and Pinkel[3] have shown that the heat-transfer coefficient q can be assumed, for practical purposes, to be independent of

[3] See N.A.C.A. Technical Report No. 488, 1934; Heat Transfer from Finned Metal Cylinders in an Airstream, by Arnold E. Biermann and Benjamin Pinkel.

[4] See N.A.C.A. Technical Note No. 429, 1932; Heat Dissipated from a Finned Cylinder at Different Fin-Plane to Airstream Angles, by Oscar W. Schey and Arnold E. Biermann.

Fig. 2—Airflow Pattern for Adjacent Fin-Surfaces

the fin width and the fin thickness. The two important factors affecting the heat-transfer coefficient are the average airspace between adjacent fin surfaces and the airspeed.

The average heat-transfer coefficients of all the cylinders tested, including both tapered and rectangular fins, are shown in Fig. 4, plotted on logarithmic coordinates against the average fin space for airspeeds from 30 to 150 m.p.h. The average deviation of the test points from the curves is 8.6 per cent. The curves for all the airspeeds have a constant slope of 0.322.

The values of q from Fig. 4 were cross-plotted in Fig. 5 against airspeed on logarithmic coordinates for a fin space of 0.175 in. The slope of the curve shows that the value of q varies with the 0.796 power of the airspeed. This exponent holds fairly well for the range of fin spacing tested.

The effect of airspeed on the surface heat-transfer coefficient for a smooth cylinder mounted with the axis both perpendicular and parallel to the airflow is also shown in Fig. 5. With the axis of the cylinder parallel to the airflow, the range of airspeeds could be extended to 215 m.p.h. The values of the slopes of the curves are 0.78 and 0.66 for the parallel and perpendicular positions, respectively. The transition point at which the airflow changes from laminar to turbulent for the case of the axis perpendicular to the airflow is clearly defined on these curves and occurs at a Reynolds Number, based on the diameter of the cylinder, of 109,000.

Calculation of Heat Flow through Fins

The development of an equation connecting the heat flow from the surface of the fins with the heat flow through the fins from the cylinder wall to the fin surfaces has been reported[3]. The equation giving the heat dissipated per square inch of cylinder-wall area per degree of average cylinder-wall temperature is:

$$U = \frac{q}{s+t}\left[\frac{2}{a}\left(1+\frac{w}{2R_b}\right)\tanh a\,w_1 + s_b\right] \qquad (1)$$

in which

U = Heat dissipated per hour per degree per square inch of cylinder-wall surface per degree of average temperature difference between the cylinder wall and the air
s = Average airspace between the fins, inches
t = Average thickness of the fins, inches
s_b = Space between the fins at the root, inches
t_t = Thickness of the fins at the tip, inches
k = Thermal conductivity of the fins, B.t.u. per in. per deg. fahr. per hr.
w = Fin width, inches
w_1 = Fin width corrected for tip loss, inches
$w_1 = w + (t_t / 2)$
R_b = Radius from the center of the cylinder to the fin root, inches
q = Average surface heat-transfer coefficient, B.t.u. per hr. per sq. in. per deg. fahr.
$a = \sqrt{(2q / kt)}$

Equation (1), together with the known values of q, permits the calculation of the total heat dissipated from the finned cylinder as a function of the average cylinder-wall temperature.

The experimental values of the overall heat-transfer coefficient are compared in Fig. 6 with the values calculated from Equation (1) for airspeeds from 30 to 150 m.p.h. The comparison has been made for all the finned cylinders tested and the agreement has been found to be good for both tapered and rectangular fins.

Design of Cylinder Fins

It is possible to design a large variety of fins of different dimensions that will dissipate the same quantity of heat from a cylinder for the same average cylinder temperature. The aircraft-engine designer is interested, however, in the fin design giving the specified heat dissipation with the use of the

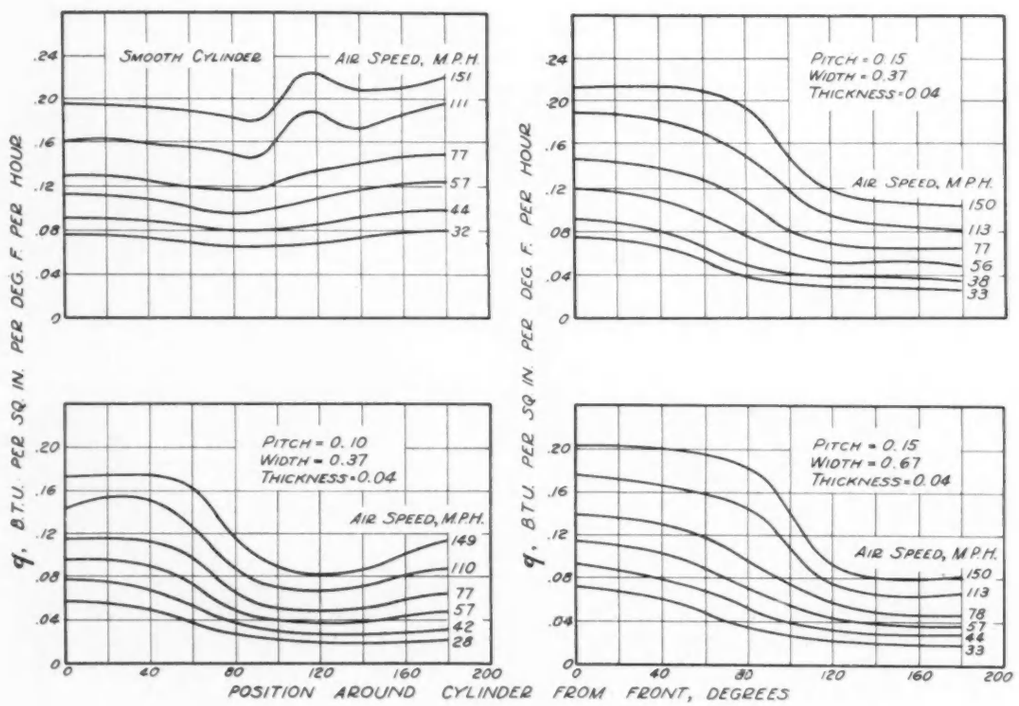


Fig. 3—Effect of Position around the Cylinder on the Heat-Transfer Coefficient q

minimum of material. He is also interested in the determination of the fin design that will dissipate the maximum quantity of heat for a given weight of material.

The pitch, width and thickness, of the fins on air-cooled engines have been limited to dimensions that could be readily cast and machined. A recent air-cooled engine has several cast-aluminum fins on the cylinder head having a pitch of 0.25 in., an average thickness of 0.085 in. and a width of 1.56 in. Realizing that several years ago the casting of fins with these dimensions would have been impossible, proves the foundry and machining limitations to be continually changing; the job that is considered impossible today may be the standard practice of tomorrow. The thickness and spacing of the fins will, however, always be limited by practical considerations of manufacture.

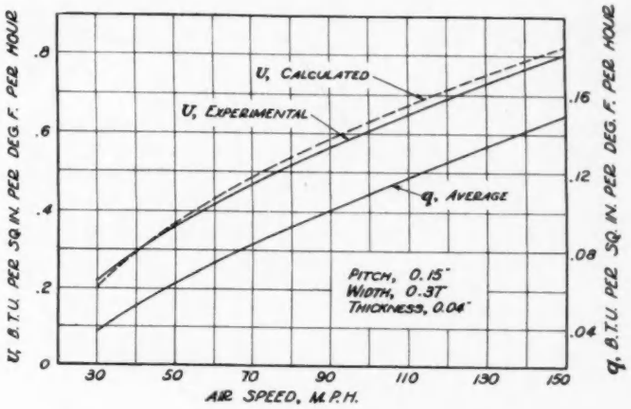


Fig. 6—Comparison of Calculated and Experimental Values of the Overall Heat-Transfer Coefficient

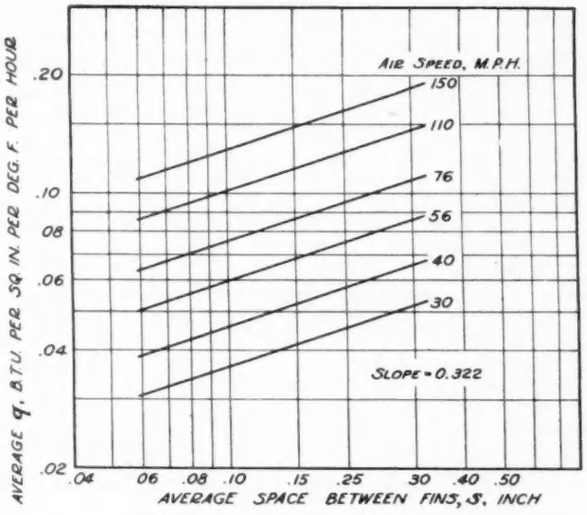


Fig. 4 *(above)*—Variation of Average q with the Average Fin-Space for All Cylinders Tested

Formulas have been developed[3] from Equation (1) for determining the dimensions of the fin that will dissipate the largest amount of heat for a given expenditure of metal and still be consistent with good manufacturing practice. These same expressions will also give the dimensions of the lightest fin construction for a given quantity of heat to be dissipated. The fins having these dimensions will be designated "optimum fins."

Fig. 7 shows a plot of the optimum fin-dimensions, the heat dissipated U, and the criterion of the weight economy U/M for various values of the thickness specified as a minimum.

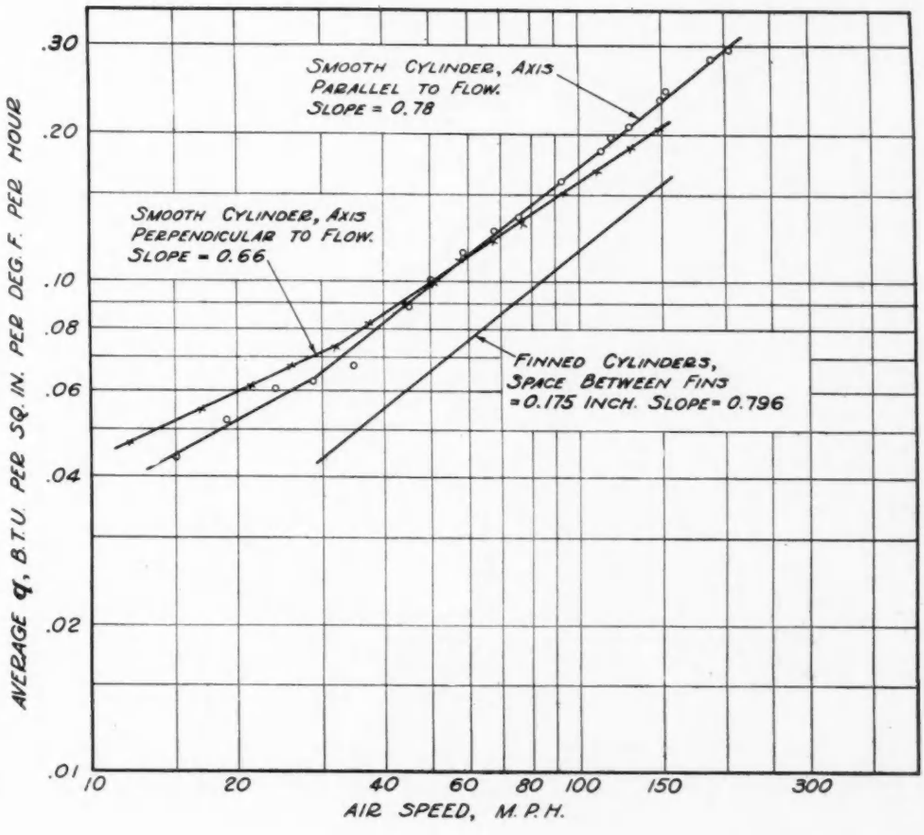

SMOOTH CYLINDER, AXIS PARALLEL TO FLOW. SLOPE = 0.78
SMOOTH CYLINDER, AXIS PERPENDICULAR TO FLOW. SLOPE = 0.66
FINNED CYLINDERS, SPACE BETWEEN FINS = 0.175 INCH. SLOPE = 0.796
AVERAGE q, B.T.U. PER SQ. IN. PER DEG. F. PER HOUR
AIR SPEED, M.P.H.

Fig. 5—Effect of Airspeed on the Average Surface Heat Transfer Coefficient

In these plots, M is the volume of fin per unit of cylinder-wall area. Fig. 8 shows a similar plot when the minimum spacing is specified as the controlling dimension. These figures have been developed for a steel cylinder having an outside diameter of 4.66 in. in an airstream of 76 m.p.h. Similar plots are being made for steel and aluminum fins at airspeeds from 50 to 175 m.p.h. and a range of cylinder diameters from 4.5 to 6.0 in.

An examination of Figs. 7 and 8 shows that the lower the specified minimum thickness and spacing is, the lighter are the fins that can be used to dissipate a given amount of heat. The curves also indicate that, for a specified fin-spacing, the quantity of heat dissipated can be increased by increasing the fin thickness and width, this increase being obtained, however, at a sacrifice in weight economy. For a specified thickness, the heat output can be increased by decreasing the spacing and slightly increasing the width.

[5] See N. A. C. A. Technical Note, 1934; The Effect of Deflectors on the Temperature Distribution and Heat-Transfer Coefficient of Finned Cylinders, by Oscar W. Schey and V. G. Rollin.

Design of Cylinder Baffles

The results presented thus far have all been obtained with the cylinder in a free airstream. At present, all high-output radial air-cooled engines use either ring or N.A.C.A. cowlings to decrease the engine drag. Satisfactory engine cooling is obtained with a reduced frontal opening by using some form of deflector or baffle around the cylinder to assure that all the air flowing through the cowling is used to dissipate heat from the cylinder fins.

The N.A.C.A. has recently completed an investigation of several types of cylinder baffles[5]. The investigation was conducted with electrically heated cylinders having fin pitches of 0.25 and 0.31 in. and a range of fin widths from 0.6 in. to 1.22

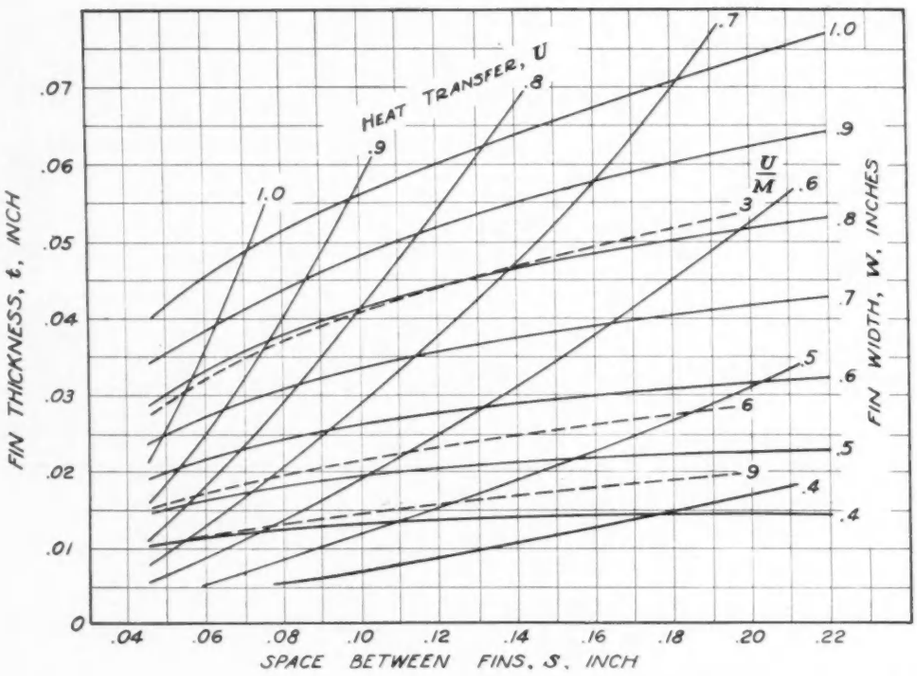


Fig. 7 *(Left)*—Optimum Fin Dimensions for the Condition in Which the Fin Thickness Is the Specified Governing Dimension

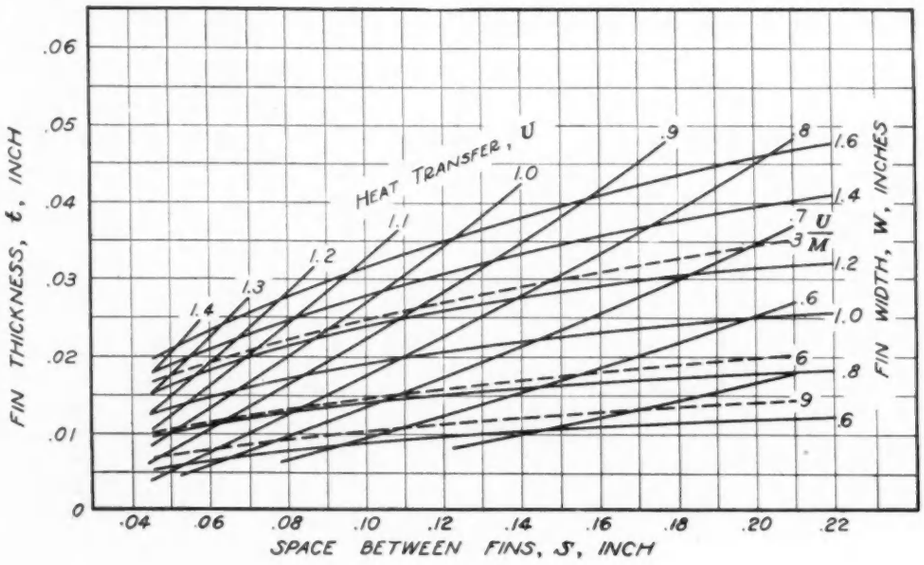


Fig. 8 *(Right)*—Optimum Fin Dimensions for the Condition in Which the Fin Space Is the Specified Governing Dimension

in. The arrangements tested included flat plates mounted at various angles with respect to the axis of the cylinder, shell baffles at different distances from the cylinder fins, and integral baffles formed by welding sheet-metal strips between the tips of the fins.

The temperature distribution around an air-cooled cylinder at a heat input of 101 B.t.u. per sq. in. of outer cylinder-wall surface and an air speed of 56 m.p.h. is shown in Fig. 9. The increase in temperature at the front of the cylinder and at the 120-deg. position from the front is in agreement with the flow pattern shown in Fig. 2. The average temperature-difference between the air and the base temperature θ_b is 236 deg. fahr. The average temperature-difference between the fin surface and the air θ is 198.8 deg. fahr.

The shell type of cylinder baffle at present is the most popular. This baffle completely surrounds the cylinder barrel with a relatively large opening in the front and a small opening in the rear. The effect of the entrance and exit angle of the baffle, the position of the baffle in respect to the cylinder, the relation of the flow area between the fins to the exit area, and the extension of the baffle at the rear of the cylinder, have been the subject of separate tests. The results indicate that, for best performance, the baffle should fit tightly against the cylinder fins, the entrance angle should be 145 deg., the ratio of the exit area to the free-flow area between the fins should be 1.6, and the extension at the rear should be approximately 3 in.

Fig. 10 shows the temperature distribution obtained when the finned steel-cylinder was tested with the best shell baffle. The baffle prevents the break-away of the air from the cylinder and gives a substantial reduction in the rear-cylinder temperature. The average heat-transfer coefficient in these tests varied as the airspeed to the 0.85 power. The use of the baffle increased the average heat-transfer coefficient 28.3 per cent.

Integral baffles have been previously tested[6] and the results reported to be unsatisfactory. The results presented in Fig. 11 show, however, that this baffle, as constructed and tested by the N.A.C.A., gives lower and more uniform cylinder temperatures than any of the baffles tested. It may be noted that the integral baffle reduced the temperature at the rear of the cylinder 50 deg. fahr. more than the best shell baffle. The large reduction in temperature obtained when using the integral baffle is attributed to the use of the surface of the deflector to dissipate heat from the cylinder fins. The average heat-transfer coefficient with the integral baffle is increased 64.6 per cent as compared with the cylinder without baffles. The difficulties in manufacturing this form of cylinder baffle may prevent its general use on aircraft engines.

[6] See 1930 Report by the Bureau of Aeronautics, Navy Department, Serial No. AEL-296; Tests of Various Types of Single-Cylinder Engines.

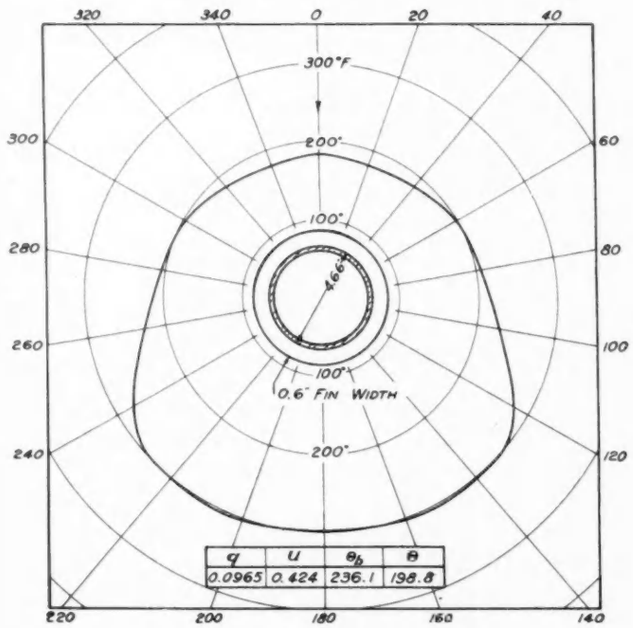


Fig. 9—Temperature Distribution around a Finned Cylinder

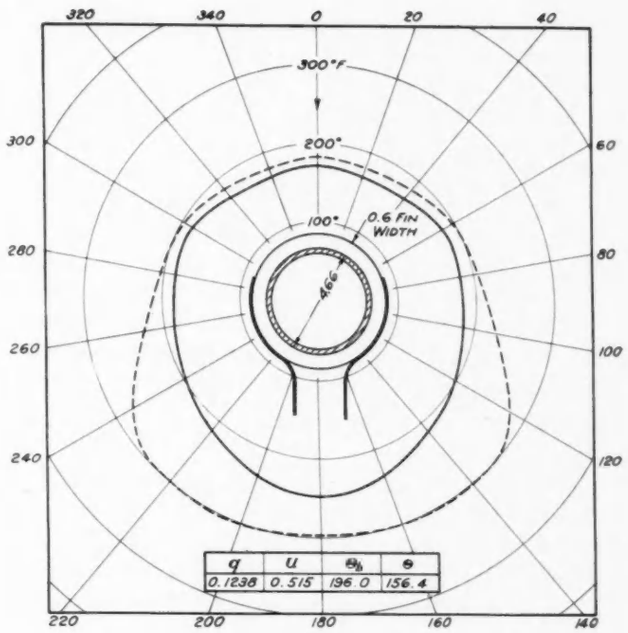


Fig. 10—Comparison of Temperature Distribution Obtained with Best Shell Baffle

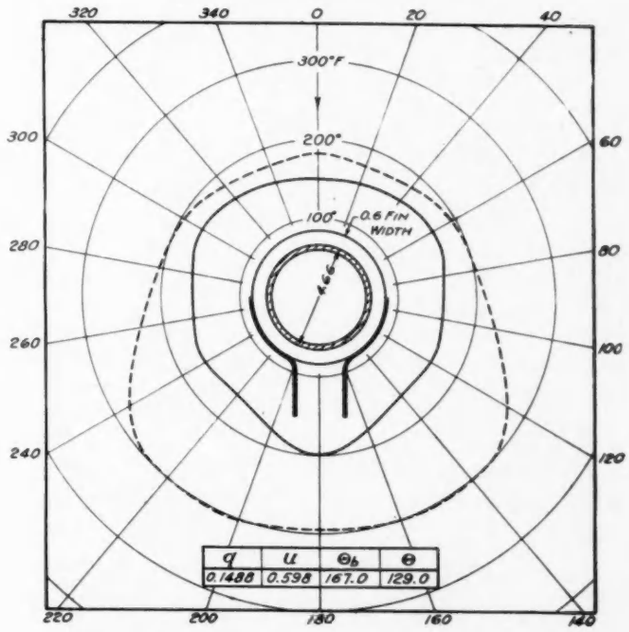


Fig. 11—Comparison of Temperature Distribution Obtained with Integral Baffle

A Suggested Index of Diesel Fuel Performance

By A. E. Becker and H. G. M. Fischer
Standard Oil Development Co.

THE ignition and detonation phases of the fuel problem were discussed in a previous paper which is cited, C.F.R.-engine data for 20 fuel oils being given, and the results were expressed in terms of the critical compression ratio. Since then, further analysis of the data indicated even better correlation between the critical compression ratio and a function of aniline point and gravity.

Since both of these physical constants are so readily determinable by simple and standardized laboratory methods, the authors propose that this function be used as a "Diesel Index" of ignition and combustion characteristics of fuel oils.

The present paper shows the correlation of this "Index" with critical compression ratio, not only by use of the data cited in the paper referred to but also for some 30 additional fuels.

AS the field for high-speed Diesel-engines expands, the question of suitable fuels and methods of rating them becomes of increased importance. That technical men are aware of this fact is clearly indicated by the increasing number of technical papers on the subject and by the recent organization of the Volunteer Group for C. I. Fuel Research, one immediate object being the development of a method for measuring the ignition quality of Diesel fuel with the expectation that it can later be recommended for universal adoption as a standard method.

The ignition and detonation phases of the fuel problem were discussed last year in a paper[1] by Becker and Stacey. It was pointed out that it is hardly to be expected that ignition quality of Diesel fuel will ever attain the importance of anti-knock quality of gasoline, although the significance of the former as applied to Diesel fuels parallels that of the latter for gasolines.

In a spark-ignition engine the efficiency improves as the compression ratio is raised, but with a resulting increase in the tendency to knock. In a compression-ignition engine, this tendency of a fuel to detonate is decreased as the efficiency is increased by raising the compression ratio. Thus, ignition quality of the fuel used determines the lower rather than the upper limit of efficiency of a Diesel-type engine. Accordingly, it is hardly necessary that an ignition-quality test be as accurate as that for determining the octane-number rating of a gasoline.

C.F.R.-engine data for 20 fuel oils were given in the aforementioned paper[1], results having been expressed in terms of the critical compression ratio (C.C.R.) obtained by the method suggested by Pope and Murdock[2]. It was shown that these engine data correlate fairly well with such physical and chemical constants as aniline point, spontaneous-ignition temperature (S.I.T.) and iodine number. Based on these results it was our opinion that it should be possible to develop either S.I.T., C.C.R., or measurement of "delay," as a labora-

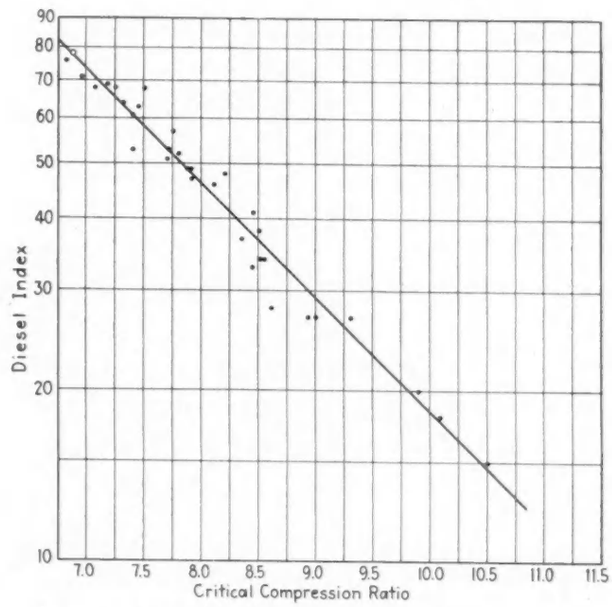


Fig. 1—Curve Showing That C.C.R. Varies as the Logarithm of the Diesel Index

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

[1] See *Proceedings of the World Petroleum Congress*, London, July, 1933: Measurement of Ignition Qualities of Diesel Fuels, by A. E. Becker and H. R. Stacey.

[2] See S.A.E. JOURNAL, March, 1932, p. 136; Compression-Ignition Characteristics of Injection-Engine Fuels, by A. W. Pope, Jr., and J. A. Murdock.

Table 1—Data for Various Fuels

Fuel No.	A. P. I. Gravity	Aniline Point, Deg. Fahr.	Dispersion at 100 Deg. Fahr.	S. I. T.	Ignition Index	Diesel Index	C. C. R.
14	20.9	77[a]	....	514	..	16	10.70
15	20.9	73	1.68	508	25	15	10.50
16	21.5	93[a]	....	505	..	20	9.90
1	22.2	120[a]	....	496	..	27	9.30
1-*A*	33.8	113	1.22	494	51	38	8.50
1-*B*	23.6	115	1.47	502	36	27	9.00
1-*C*	16.3	...	....	510	..	..	10.00
10	25.0	...	....	491	..	..	8.95
4	30.4	112	1.30	495	46	34	8.55
3	26.8	152[a]	....	494	..	41	8.45
11	30.4	...	....	487	..	..	8.35
2	29.9	152	....	486	..	46	8.12
5	38.8	122	1.06	485	60	47	7.92
18	32.6	149	1.09	472	65	49	7.90
17	35.2	147	1.15	...	63	52	7.80
9	41.7	136	1.00	484	68	57	7.75
19	34.9	151	1.01	476	65	53	7.70
20	33.3	154	....	474	..	51	7.70
8	44.8	153	0.95	474	80	68	7.50
12	39.3	160	0.93	474	80	63	7.45
7	34.7	176+	1.02	464	84	61	7.40
13	36.4	170	....	472	..	64	7.32
6	37.8	176+	....	462	..	68	7.25

[a]Approximate

tory test that would correlate satisfactorily with field service and that the adoption of a suitable scale or index should hasten this development and avoid confusion in the industry.

Since that time, H. G. M. Fischer has found by further analysis of the data that there is even better correlation between C.C.R. and a function of aniline point and gravity. Both of these physical constants are so readily determinable by simple and standardized laboratory methods that we are proposing this function be used as a "Diesel Index" of ignition and combustion characteristics of fuel oils, as defined by the equation:

$$\text{Diesel Index} = \frac{\text{Aniline Point (Deg. Fahr.)} \times \text{A.P.I. Gravity}}{100} \quad (1)$$

[3] See S.A.E. JOURNAL, July, 1931, p. 3; Fuels for Automotive Diesel Engines, by W. H. Butler, (abstract).

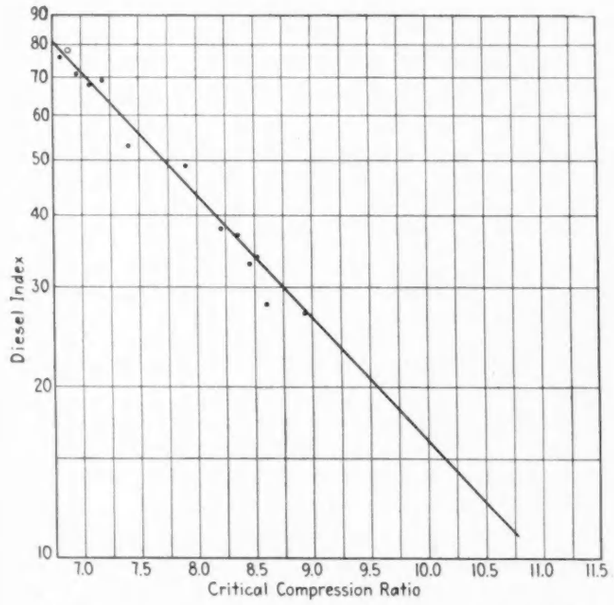


Fig. 2—Curve Resulting from Data Taken from Table 3

The present paper shows the correlation of this "Index" with C.C.R., not only by use of the data given in the paper[1] mentioned but also for some 30 additional fuels. These were collected from widely different sources and are representative of both cracked and straight-run products from asphaltic, paraffinic, and mixed-base crudes. To permit utilization of previous field data, the degree of correlation of the suggested "Diesel Index" with S.I.T. is shown also, as well as with a previously used "Ignition Index" based on somewhat different physical constants, as discussed in a paper by W. H. Butler[3].

The A.P.I. gravity, aniline point, S.I.T., and C.C.R. data of Table 1 are taken directly from the paper by Becker and Stacey[1]. For convenience in comparing the various indices, the fuels are listed in the order of decreasing C.C.R. Diesel Indices, as calculated from the gravities and aniline points. More complete inspection data for these fuels are included in

Table 2—Data on the Additional Fuels Tested

Fuel No.	A. P. I. Gravity	Aniline Point Deg. Fahr.	Diesel Index	C.C.R.
21	42.6	182	78	7.08
22	35.8	169	60	7.25
23	46.9	158	74	7.35
24	31.2	171	53	7.40
25	38.9	151	58	7.52
26	33.4	154	51	7.52
27	32.2	155	50	7.55
28	33.4	141	47	7.71
29	38.0	133	50	7.73
30	32.4	139	45	7.90
31	34.0	146	50	7.94
32	38.1	126	48	7.95
33	33.4	131	44	8.02
34	30.4	147	45	8.07
35	28.4	134	38	8.25
36	33.0	111	37	8.25
37	30.5	107	33	8.80
38	24.4	109	27	8.89
39	29.8	98	29	8.95
40	29.5	94	28	9.05
41	47.7	132	63	8.07

Table 3—Data on a Wide Range of Fuel Types

Fuel	A. P. I. Gravity	Aniline Point, Deg. Fahr.	S. I. T.	Diesel Index	C. C. R.
42	36.1	212	...	76	6.82
A	41.2	172	461	71	6.95
H (6)	37.8	176+	462	68	7.07
E	41.7	166	468	69	7.18
24	31.2	171	...	53	7.40
G	37.8	129	488	49	7.88
J	29.4	131	490	38	8.21
B	33.6	111	488	37	8.35
C	28.2	117	489	33	8.45
43	31.0	109	...	34	8.52
44	24.8	113	...	28	8.61
I (1)	22.2	120	496	27	8.93
D	20.0	92	495	18	10.09
F	17.4	...	516	...	11.03
Cetene	48.6	161	456	78	6.87
Alpha-methyl-naphthalene	....	Too dark	Above 700	...	Very high
Normal heptane	74.1	158	468	117	7.35
Iso-octane	71.6	181	602	130	16.5

Table 2 of the original paper. The C.C.R. data were obtained with the following engine-testing conditions:

Engine Speed, r.p.m.	600
Jacket Temperature, deg. fahr.	210
Air-Inlet Temperature, deg. fahr.	100
Oil Temperature, deg. fahr.	130
Injection Pressure, lb. per sq. in.	1420
Injection-Angle Advance, deg.	12
Total Time of Injections, sec.	3
Fuel per Injection, cc.	0.023

The same testing conditions were used for the additional fuels listed in Table 2 of the present paper. By maintaining these conditions it was possible to obtain results which were reproducible within ± 0.1 of a compression ratio.

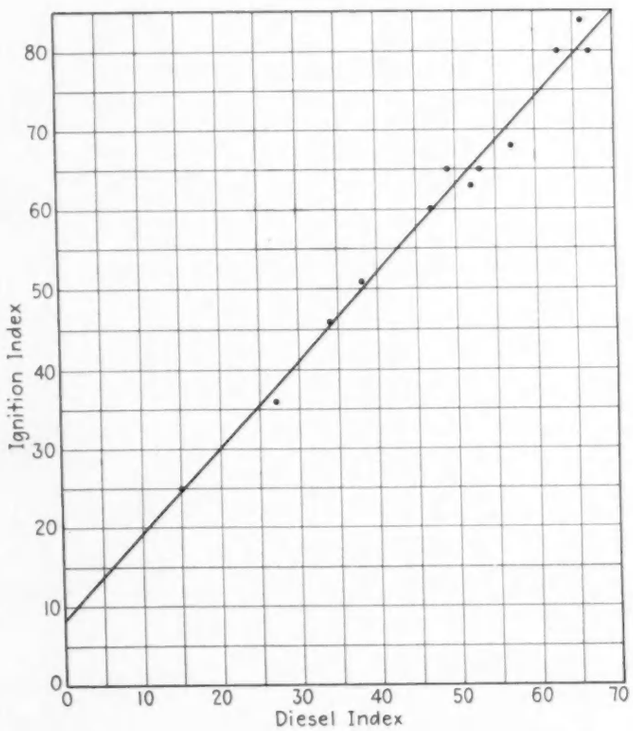


Fig. 3—Curve Showing the Agreement between the Diesel Index and the Ignition Index

It will be noted that the Diesel Index decreases as C.C.R. increases. The same data, except for fuel 41, are shown in Fig. 1, from which it will be seen that C.C.R. varies as the logarithm of the Diesel Index. The data agree with the curve within ± 0.25 of a compression ratio.

Perhaps even better correlation would be obtained if it were possible to maintain engine-testing conditions more accurately. This is indicated by the fact that a straight line of somewhat smaller intercepts fits the data of Fig. 2 better. These data are taken from Table 3, the C.C.Rs. having been obtained under the same engine-testing conditions as before, except that the injection pressure was raised 80 lb., to 1500 lb. per sq. in. and that the amount of fuel per injection was increased from 0.023 cc. to 0.03 cc. The lettered fuels are some which have been interchanged by members of the Volunteer Group for C.I. Fuel Research. From the inspection data furnished by these members, Table 4, it will be noted that they cover a wide range of fuel types.

In view of the sensitiveness of C.C.R. measurements to engine-testing conditions, it is to be expected that it will be difficult to obtain agreement between different engines and laboratories or even on the same engine from one day to another unless testing technique is very carefully standardized, just as has been done in the method for determining the octane number of gasoline. The same thing is likely to be true of the engine "delay" method. On the other hand, the Diesel Index depends upon two laboratory tests so easily made that even a novice can obtain reproducible results, certainly for those fuels of high ignition-quality.

The Diesel-Index scale is also of a satisfactory range for commercial usage. From Table 3 it will be noted that cetene has a value of about 78 and, from Fig. 2, that this value (indicated by a circle) is in close agreement with the C.C.R. curve. Similar data for alpha-methyl-naphthalene are difficult to obtain, but there is every reason to expect that mixtures of these two pure compounds will act normally. We have just found that a mixture of 35 per cent cetene and 65 per cent alpha-methyl-naphthalene has a C.C.R. of 9.9, a gravity of 20.7, an aniline point of 64 deg. fahr., and a Diesel Index of 13, which values are in fair agreement with the curve in Fig. 2.

On the other hand, both iso-octane and normal heptane have considerably higher C.C.R. ratings than would be expected from their Diesel Indices. But these are in the gaso-

line range and of no immediate concern in the Diesel-fuel picture. A similar result, although considerably less pronounced, was obtained for fuel 41, Table 2. However, this fuel is a heavy naphtha and therefore not representative of Diesel-type fuels. To date we have not encountered any fuel oil for which the C.C.R. cannot be closely predicted from its Diesel Index.

As stated, some field work has been correlated with the "Ignition Index" proposed and used by W. H. Butler, as defined by the following formula:

$$I = \frac{(1 - G) \times G \times T}{100\ D} \qquad (2)$$

in which

I = Ignition Index
G = Specific Gravity at 100 Deg. Fahr.
T = Average Boiling Point, Deg. Fahr.
D = Dispersion at 100 Deg. Fahr.

Accordingly, we obtained the dispersion values for 12 fuels with an Abbe refractometer and using the red end of the spectrum. These are shown in Table 1. Corresponding Ignition Indices are given therein also, the average boiling-points required in the formula having been calculated from the distillation data by taking an average of the temperature for the 5, 15, etc., to 95 per cent points.

The agreement between the Diesel Index and this Ignition Index is extremely good as shown by Fig. 3, the latter differing from the former substantially by a constant value. As both of these indices are measures of parafinicity, it is quite likely that there are possibilities in other yardsticks such as Viscosity Index and Viscosity Gravity Constant. We prefer Diesel Index to Ignition Index because it does not require the use of special equipment not usually available in routine laboratories. Any field work that has been based on the latter is readily interpretable in terms of the former through Fig. 3.

Spontaneous-ignition temperatures (S.I.T.) are given for a number of the fuels used in this work, in Table 1 and Table 3. These were determined in a bomb apparatus of the Moore type under the following conditions:

Oxygen Flow	100 cc. per min.
Drop Size	2-mm. diameter
Rate of Temperature Rise	1 deg. fahr. per min.

All readings were taken with the temperature rising, 1 min. being allowed between trial drops of fuel for the oxygen to purge the bomb. Results were reproducible within ± 2 deg. fahr.

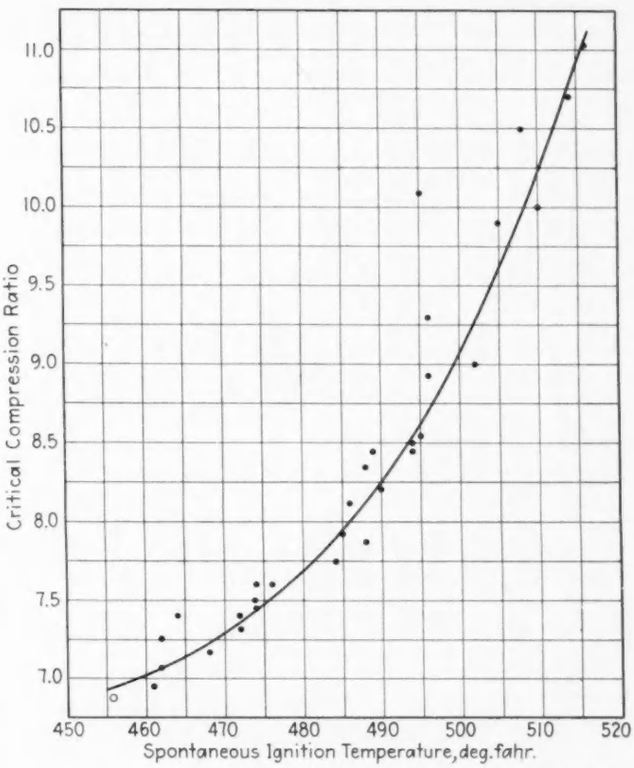


Fig. 4—Curve Showing the Relation between C.C.R. and S.I.T. for the Fuels Used in This Work

The relation between C.C.R. and S.I.T. for these fuels is shown by the curve in Fig. 4. Although not perfect, it will be noted that as one increases the other also increases. Fig. 5 shows a straight line relation between S.I.T. and Diesel Index. As the variation from the correlation line in terms of S.I.T. is about the same as that for the C.C.R.-S.I.T. curve of Fig. 4, the inference is rather strong that these variations are in large measure due to experimental errors in determining S.I.T. The use of S.I.T. as an index has the further dis-

Table 4—Inspection Data Furnished by Members of the Volunteer Group for C. I. Fuel Research

Fuel	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*
Gravity	40.9	33.7	28.0	19.9	40.0	18.0	37.5	37.8	22.2	29.1
Saybolt Viscosity at 100 deg. Fahr.	37	33	39	57	38	38	...	42	52	39
Flash, deg. Fahr.	190	190	190	205	...	...	...	260	174	...
Fire, deg. Fahr.	210	210	215	230	...	...	...	...	...	...
I. B. P., deg. Fahr.	364	397	386	340	420	420	344	510	...	404
Per Cent										
10	451	431	452	443	...	...	389	540	456	459
20	482	439	473	457	...	...	407	546	502	476
30	507	446	492	470	...	...	420	552	544	491
40	527	453	509	483	...	...	437	558	586	510
50	545	461	527	504	500	500	451	566	608	531
60	563	470	547	535	...	...	469	574	644	552
70	582	481	569	579	...	...	488	585	663	579
80	...	500	600	...	...	...	508	599	688	611
90	...	525	...	...	...	...	...	...	...	...
F. B. P., deg. Fahr.	...	608	...	...	580	580	588	682	Above 700	733
Residue, per cent	...	1.0	...	...	...	...	1.25	2.0	14.0	...
Stock	Straight Run	Cracked	Cracked	Cracked	Straight Run	Cracked	...	Straight Run	Cracked	Straight Run

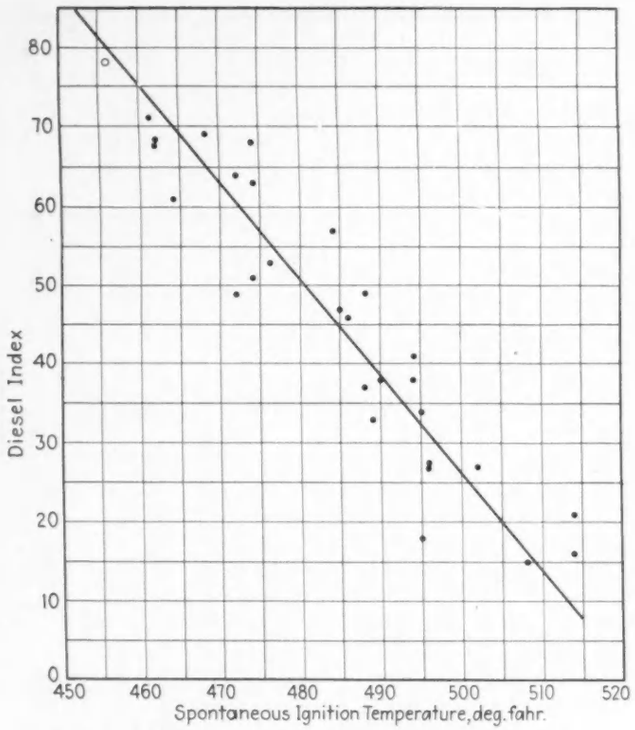


Fig. 5—Illustration of the Straight-Line Relation between S.I.T. and Diesel Index

advantages that it is a special test not usually made and that the temperature range between a good and a poor oil is small compared with the experimental errors involved.

It is therefore our suggestion that Diesel Index be used as a temporary standard for ignition quality of fuel oil. One advantage, over those of other proposed indices, is that it depends upon two physical constants, the determination of each being only a matter of a few minutes' work. One of these, gravity, is customarily taken on all tests samples; the other, aniline point, is determined by heating equal proportions by volume of aniline and the sample in a jacketed test-tube to a clear solution and then noting the temperature at which turbidity appears as the mixture is cooled.

Further field data will determine the suitability of the Diesel Index as a permanent standard of Diesel-fuel performance. It is certainly promising in that it would permit engine methods and other yardsticks requiring special apparatus to be dispensed with, or at least limited to research and control laboratories.

Discussion

Engine Measurements Made by Two Methods

—J. R. MacGregor

Research Engineer, Standard Oil Co. (Calif.)

THE use of our data by Dr. Becker in the preparation of his paper has come to me as somewhat of a surprise. However, since the subject has been brought up, a brief discussion of the methods used in our laboratories seems in order.

All of our tests are made in the Waukesha conversion of the regular C.F.R. detonation test-engine with the exception of certain modifications which will be described later. In these tests the engine is used as a dynamic bomb in which fuel is injected under control of an electric timer for a predetermined number of successive power strokes followed by a predetermined number of scavenging strokes. The electric timer, by which this control of injection is obtained, consists of a system of breaker points and commutator driven from the crankshaft by a take-off installed in the normal tachometer-drive opening. By adjustment of the period of contact of the breaker points, the number of successive injection strokes is controlled, while suitable positioning of the selector switch in conjunction with the six-brush commutator provides control of the ratio between injecting and non-injecting strokes. This device was originally designed for research purposes and provided means for injecting during one or more strokes followed by any desired number of scavenging strokes from 1 to 60. However, experience has indicated that a cycle consisting of two injection strokes followed by 28 scavenging strokes has provided very satisfactory reproducibility and freedom from engine fouling. This cycle was the one used in obtaining the data which Dr. Becker has used in his paper.

Experience has indicated a considerable amount of difficulty when testing by the Pope and Murdock method, due to the liberation of gases when the exhaust listening-method was used for the determination of the presence of combustion. We have overcome this difficulty by the installation of a modified Midgley indicator which provides a means for determining if combustion takes place and at what part of the power stroke it occurs. The engine fouling has been materially reduced by changing the cone angle of the injection nozzle from the original 30 deg. to one of 8 deg., remachining the head to position the end of the injection nozzle flush with the cylinder wall and depressing the angle of injection 2½ in. from the horizontal. These alterations have practically eliminated the deposition of fuel on the relatively cool piston and cylinder-wall areas.

The other engine conditions used in making our test are essentially those of Pope and Murdock, in which the engine is driven by the synchronous motor at 600 r.p.m., with jacket-water temperature maintained constant at 210 deg. fahr. The intake-air temperature is maintained at 100 deg. fahr. and the oil temperature is held at its equilibrium point of about 130 deg. fahr. The injection advance is maintained at 12 deg. before top center and the injection valve is set to open at 2000 lb. per sq. in. Test fuels are bracketed between blends of reference fuels until a match is obtained, and the results are reported in terms of cetene number as well as critical compression-ratio.

The method we use in determining the ignition lag also uses the Waukesha engine with the injection-nozzle modifications mentioned previously but without the use of the electric timer. The engine is operated by the synchronous motor at 1200 r.p.m. at a constant compression ratio of 14 to 1 with engine temperature maintained constant as in the starting test.

The timing of injection is noted by the regular neon-flash protractor, while the point in the cycle at which ignition occurs is noted by a second discharge through the same protractor but controlled by the opening of the contacts of a modified bouncing-pin. This bouncing-pin is so constructed that it is independent of actual cylinder pressures but de-

pendent upon the rate of rise of pressure. By this means it is possible to determine the ignition lag by noting the angular difference on the protractor between the flash at the opening of the injection nozzle and the flash produced by the opening of the bouncing-pin contacts.

In testing, the unknown fuel is run continuously in the engine and the injection timing altered until the ignition lag becomes a minimum. At this point, the ignition lag and injection timing are noted and recorded. Reference fuels are then run in a similar manner until a match is obtained, and the values are reported in cetene numbers.

It will be noted in the curves presented by Dr. Becker that rather satisfactory relations are shown between his Diesel Index and both our starting-test and ignition-lag-test results. However, it will be noted also that there is a considerable difference in the slope of the curves, which indicates the influence of engine-test conditions upon the results obtained. We feel that the data obtained in our starting tests are quite reproducible and afford a satisfactory criterion of the combustion characteristics of Diesel fuels under the particular engine conditions used. The data obtained in the starting tests, however, are not quite so reproducible due, probably, to the small spread in ignition lag between the fuels of widely differing character.

In closing, I wish to warn against the early adoption of any laboratory-test method until adequate correlation with service behavior of a variety of fuels has been obtained. The influence of two sets of engine conditions has been well demonstrated by our data as used by Dr. Becker.

Diesel Index and C.C.R. Results Correlation

—A. E. Becker

Standard Oil Development Co.

AFTER completion of the paper entitled "A Suggested Index of Diesel-Fuel Performance" I received J. R. MacGregor's letter of June 8, 1934, to the members of the Volunteer Group for C.I. Fuel Research, reporting results obtained in the laboratories of the Standard Oil Co. (Calif.) for those samples of Table 3 designated by the letters *A* to *I*. He has very kindly consented to my showing of these results at this time. The data are particularly interesting because they have been obtained in a different laboratory when using engine-testing conditions which differ materially from those which were used in obtaining the data given in the paper. I have termed the following discussion the Correlation of the Diesel Index with C.C.R. Results Obtained in the Laboratories of the Standard Oil Co. (Calif.).

Mr. MacGregor made engine measurements by two methods which he designates as the "Starting Method" and the "Ignition-Lag Method" and concerning which he has given details in his preceding discussion. In the first he determined C.C.R. and cetene numbers; in the second, minimum ignition-lag and cetene numbers. His results by the Starting Method are shown in Fig. *A*, cetene numbers being plotted against the corresponding C.C.R. values. I have also plotted his C.C.R. data against the Diesel Indices as taken from Table 3 of the paper. It will be noted that the correlation of C.C.R. with the Diesel Index is equally as good as with the cetene number, both being excellent, which speaks well for

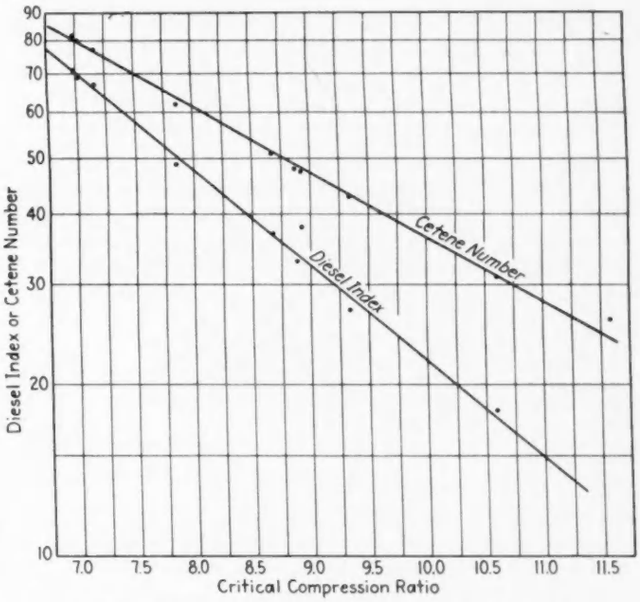


Fig. *A*—(Becker Discussion) Curve Resulting from Plotting Cetene Numbers against the Corresponding C.C.R. Values

this Starting Method as an engine test. It will be noted that the difference between the cetene number and the Diesel Index increases slightly as C.C.R. increases.

The slope of the Diesel-Index line varies materially from those of the correlation lines shown in Figs. 1 and 2 of the paper. This very definitely shows the influence which engine-testing conditions have on C.C.R. data and serves to stress the value of the Diesel Index as a measure of quality. The latter depends upon two tests which can much more readily be rigidly controlled than engine tests can, and there-

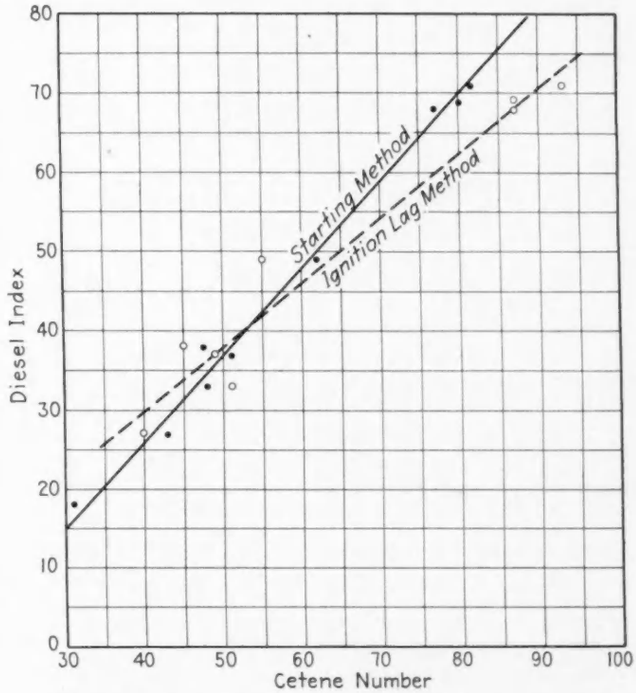


Fig. *B*—(Becker Discussion) Curve Resulting from Plotting Cetene Numbers against Diesel Indices

fore more easily duplicated from one laboratory to another.

This same point is also clearly indicated by the data shown in Fig. *B*, where Mr. MacGregor's cetene numbers by the Starting and the Ignition-Lag methods, respectively, are plotted against the Diesel Indices as taken from Table 3. Whereas the cetene numbers obtained by the two methods differ materially, it will be noted that the Diesel Index can be used to predict the cetene number that would be obtained by either method. This means that the Diesel Index is also a criterion of detonation as well as of ignition quality.

Full-Scale Engine-Tests Of Diversified Fuels

—C. H. Baxley

Staff Engineer, Sinclair Refining Co.

IN the development of any problem, it is always beneficial to be able to relate laboratory-test data with results obtained under service conditions. That such correlation is particularly important in fuel research has been amply demonstrated in the work of the Detonation Subcommittee of the Cooperative Fuel Research Steering Committee, and it is reasonable to suppose that in the development of methods of testing Diesel fuels a close relationship should be established between engine results and any proposed methods of test.

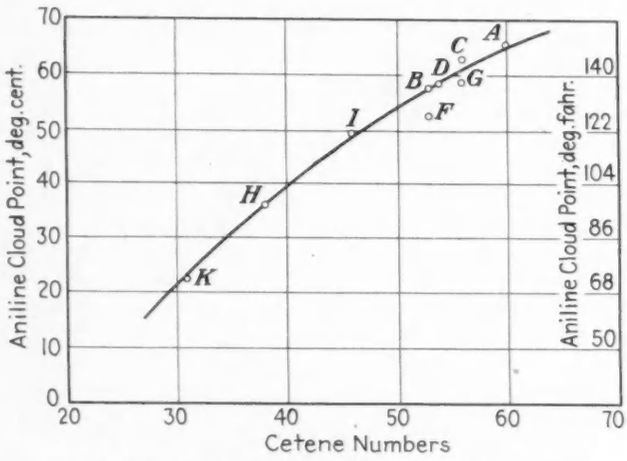


Fig. *A*—(Baxley Discussion) Aniline Cloud-Point versus Cetene Numbers

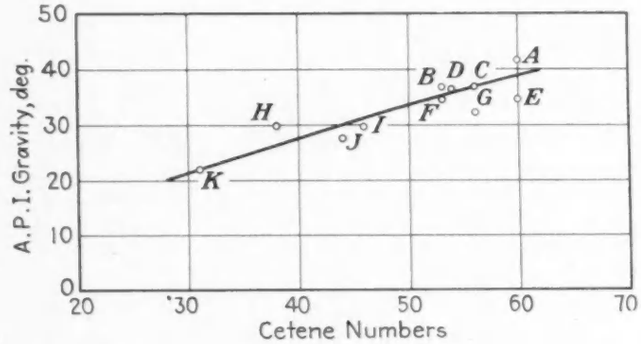


Fig. *B*—(Baxley Discussion) A.P.I. Gravity versus Cetene Numbers

Within recent months the Sinclair Refining Co. has had the opportunity of conducting some full-scale engine-tests, using a series of diversified fuels. The results obtained from these tests are quite interesting in that it is possible to draw certain conclusions regarding the fuels run, and more particularly to correlate the full-scale engine-results with several laboratory tests.

The engine used was a 600-hp. Krupp, solid-injection

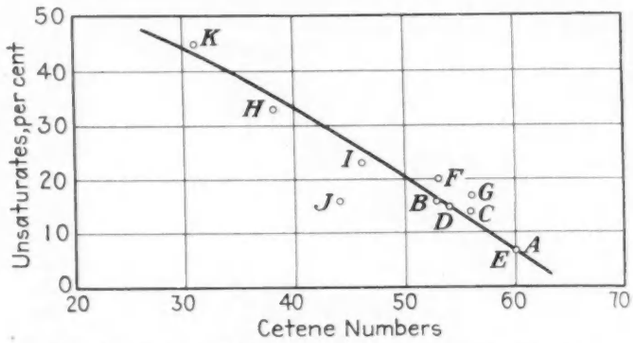


Fig. *C*—(Baxley Discussion) Percentage of Unsaturates versus Cetene Numbers

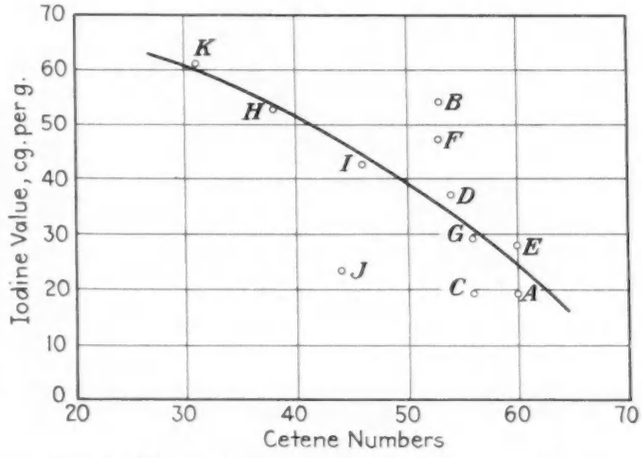


Fig. *D*—(Baxley Discussion) Iodine Value versus Cetene Numbers

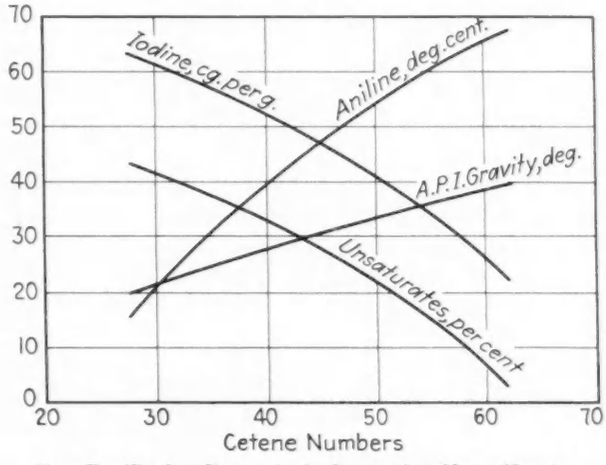


Fig. *E*—(Baxley Discussion) Composite Chart Showing Aniline Cloud-Point, A.P.I. Gravity, Iodine Value and Percentage of Unsaturates versus Cetene Numbers

Figs. *A* to *E* (Baxley Discussion) Plots of the Data in Table *A*

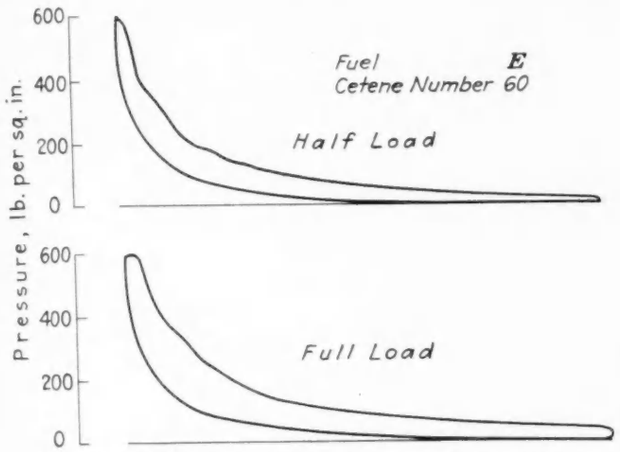


Fig. *F*—(Baxley Discussion) Pennsylvania Neutral Oil (Fuel *E*) with a 60 Cetene Number

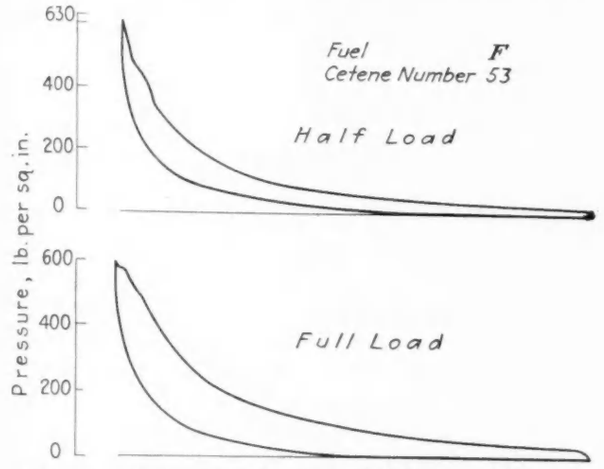


Fig. *G*—(Baxley Discussion) A Card Representative of the No. 2 Fuel Oils

Diesel, running at 360 r.p.m. The usual engine data were obtained, including indicator cards at full load and at part loads. In the laboratory, cetene numbers were determined by the Pope and Murdock method using the modified C.F.R. engine, and the regular bench tests were run for each fuel.

Table *A* gives the principal laboratory data on the fuels.

Table *A*—(Baxley Discussion) Diesel-Fuel Analyses

Fuel	Gravity, A. P. I. Deg.	Iodine, C.J./J.	Unsaturates, Per Cent	Aniline, Deg. Fahr.	Cetene No.
A	41.4	19.4	7.0	150.1	60
B	36.7	54.3	16.0	135.1	53
C	36.7	19.5	14.0	145.5	56
D	36.1	37.6	15.0	136.9	54
E	34.6	28.1	7.0	188.1	60
F	34.4	47.5	20.0	127.9	53
G	32.2	29.4	17.0	137.1	56
H	29.7	52.8	33.0	97.0	38
I	29.5	42.7	23.0	120.4	46
J	27.3	23.9	16.0	.....	44
K	21.7	61.3	45.0	72.1	31

Table *B*—(Baxley Discussion) Operating Data

Fuel	Fuel Consumption, Lb. per Kw. Hr. ½	Fuel Consumption, Lb. per Kw. Hr. Full	Maximum Pressure, Lb. per Sq. In. ½	Maximum Pressure, Lb. per Sq. In. Full	Start
B	0.574	0.524	630	600	Saturate
C	0.650	0.542	660	600	Saturate
E	0.540	0.520	600	600	Saturate
F	0.584	0.555	650	600	Saturate
G	0.597	0.564	650	600	Saturate
I	0.600	0.584	730	670	Unsaturate
J	0.594	0.565	690	670	Saturate
K	0.597	0.567	820	720	Unsaturate

All of the samples, with the exception of Fuel *E*, are commercial fuels; Fuel *A* is a No. 1 Fuel Oil; Fuels *B*, *C*, *D*, *F*, *G* and *H* are No. 2 Fuel Oils sold in various parts of the coun-

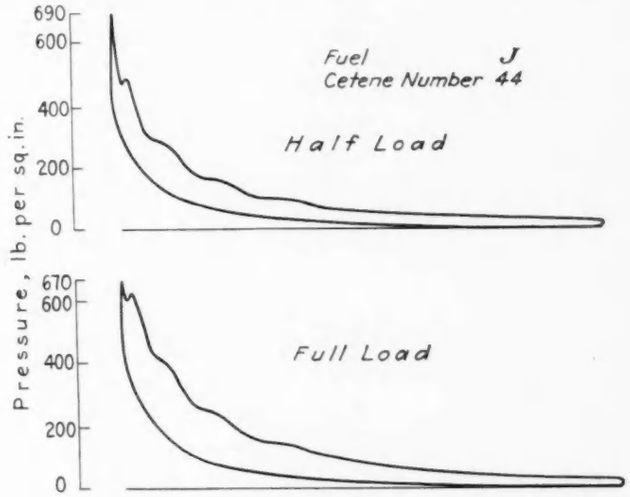


Fig. *H*—(Baxley Discussion) Indicator Card for Fuel *J*, a No. 4 Fuel Oil

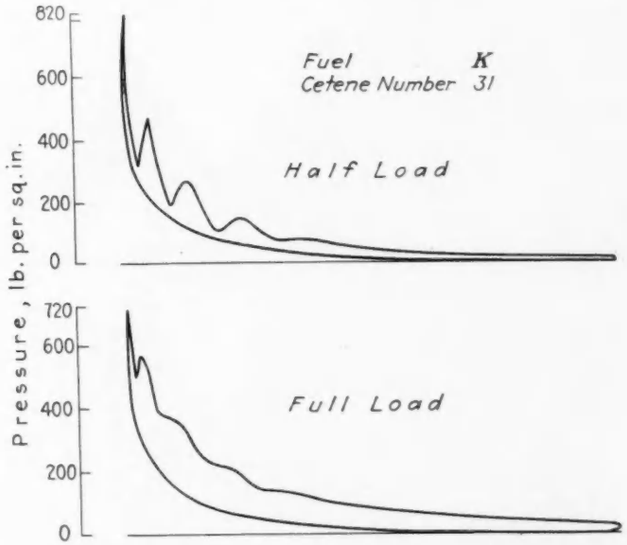


Fig. *I*—(Baxley Discussion) Indicator Card for Fuel *K*, a No. 4 Fuel Oil

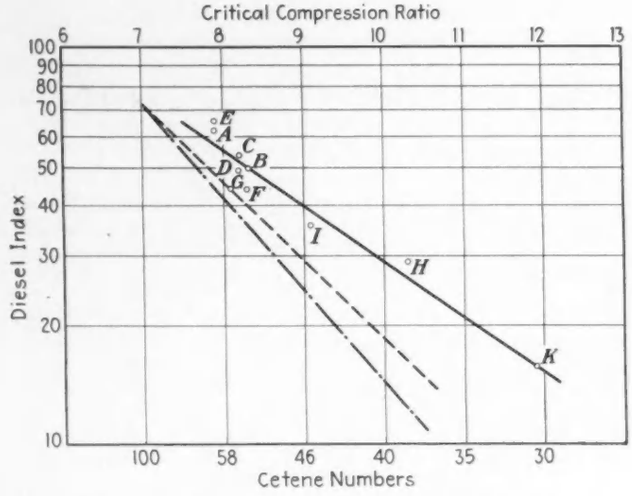


Fig. J—(Baxley Discussion) Curves Showing Diesel Indices versus Cetene Numbers and versus Critical Compression Ratios

try; Fuel *I* is classed as a No. 3 Fuel Oil, although this designation in most cases is purely nominal; and Fuels *J* and *K* are true No. 4 Fuel Oils. Fuel *E* is a non-viscous neutral oil made from Pennsylvania crude and has a Saybolt universal viscosity of 52 sec. at 100 deg. fahr. The viscosities of the other fuels, with the exception of Fuel *J*, range from 32 to 36 sec., the latter having a viscosity of 49 Saybolt universal sec. at 100 deg. fahr. The flash (P.M.) of the neutral oil is 300 deg. fahr., the others coming within the usual commercial limits. Carbon Residue, Sulphur and B. S. & W. are satisfactory in every case.

Figs. *A* to *E* inclusive are plots of the data given in Table *A*. It will be seen that the Cetene Numbers of the samples vary directly with the aniline cloud-point and the A. P. I. gravity, but conversely as the percentages of unsaturates and the iodine value. Better agreement is noted for the aniline cloud-point and gravity.

Operation data are summarized in Table *B* for the eight fuels run in the engine. Fuels *I* and *K* were noted as unsatisfactory due to poor starting, the start being made from cold in each case. The maximum pressures were scaled from the indicator cards. Fuel consumption, in pounds per kilowatt hour, indicates a preference for the higher A. P. I. gravity fuels, although this seeming advantage is eliminated when put on a cost basis.

Table *C*—(Baxley Discussion) Diesel-Index Calculation

Fuel	Gravity, A. P. I. Deg.	Analine, Deg. Fahr.	Diesel Index	Cetene No.	C.C.R.
A	41.4	150.1	62.1	60	7.9
B	36.7	135.1	49.5	53	8.3
C	36.7	145.4	53.3	56	8.2
D	36.1	136.9	49.4	54	8.2
E	34.6	188.1	65.1	60	7.9
F	34.4	127.9	43.9	53	8.3
G	32.2	137.1	44.2	56	8.1
H	29.7	97.0	28.8	38	10.3
I	29.5	120.4	35.5	46	9.1
J	27.3	.....	....	44	9.4
K	21.7	72.3	15.7	31	11.4

The indicator cards for several of the fuels are shown in Figures *F*, *G*, *H* and *I*. Fig. *J* is the Pennsylvania neutral oil (Fuel *E*) with a 60 cetene-number. The diagram shows excellent combustion, both at full load and part load, and is included here to show the optimum performance. Fig. *G* is representative of the No. 2 Fuel Oils. It also shows good combustion and pressures in line with those of the best performance. Figs. *H* and *I* are the indicator diagrams for fuels *J* and *K*, the No. 4 Fuel Oils. Fuel *J* is satisfactory. Fuel *K* is decidedly unsatisfactory, which leads to the conclusion that a cetene number of about 40 to 45 probably represents a borderline condition for this equipment.

The Diesel Index has been calculated for each of the fuels according to the method proposed by Becker and Fischer. The results are shown in Table *C* and Fig. *J*, in which the dotted line and dot-dash line are taken from Figs. 1 and 2, respectively, of the Becker and Fischer paper. Individual points are not shown for these latter curves. It will be seen that the service and laboratory data presented in this discussion are in fair agreement with those of Drs. Becker and Fischer.

It is earnestly recommended that additional service data be obtained by all those interested in the subject and in a position to do so, in order that definite relationships between actual engine performance and laboratory tests may be agreed upon.

Diesel Index Valuable In Selecting Diesel Fuels

—C. H. Schlesman

Supervisor, Automotive Dept.
Socony-Vacuum Oil Co., Inc.

The paper concerns itself with a subject which is of particularly great interest to the Society at present. At Dr. Becker's suggestion, we have calculated the Diesel Index of a considerable number of fuels which have also been tested in a Diesel engine. A correlation of combustion-delay-angle measurements with their Diesel Index and a comparison of C.C.R. values with the Diesel Index indicate that this Index is of considerable value in selecting Diesel fuels of satisfactory ignition characteristics. If the C.C.R. secured in our laboratories are plotted against the Diesel Index, in the manner suggested in the paper, it will be found that the points lie upon a straight line or within a reasonable distance from it. However, we note that a line drawn on the basis of our data falls at a considerable distance above the line shown in the paper.

The discrepancy appears to result from differences in the method of determining analine numbers and differences in the engine data reported by the two laboratories. However, we have observed that a number of Diesel fuels give conflicting ignition values when measured by the C.C.R. method and by the delay-angle measurements. Until more data can be obtained correlating the Diesel Index with engine performance, we believe that engine data, rather than chemical data, should be employed in a classification of Diesel fuels.

Because of the limitations cited, it appears inadvisable to write the Diesel Index into fuel-oil specifications at this time.

The Hesselman Low-Compression Diesel-Fuel-Burning Engine

By A. W. Pope, Jr.
Research Engineer, Waukesha Motor Co.

EIGHT different models of the Hesselman engine, ranging in bore from 3¾ to 6½ in. and in power from 25 to 170 hp., have been developed and produced since the first production engine was completed about two years ago, according to Mr. Pope, and hence it has passed the experimental stage. It is a low-compression engine that burns Diesel fuels.

Mr. Pope states that one of these production industrial engines has developed a mean effective pressure of 104 lb. per sq. in. and that a full-load economy of 0.58 lb. per b.hp-hr. has been obtained. Further, that its economy lies about half way between that of a carburetor engine and a Diesel engine.

The operation of the engine is described, suitable fuels for it are discussed, and data resulting from tests are presented. In conclusion, Mr. Pope states that the engine has provided a means for handling heavy fuels at low compressions.

THE Hesselman engine is a low-compression engine which burns Diesel fuels. Injection of the fuel is similar to the Diesel method, but ignition is accomplished by means of a spark plug instead of by the heat of compression.

Before describing the details of the Hesselman engine, it is well to point out the fact that it was invented and developed by a leading European Diesel engineer, K. J. E. Hesselman, an authority on conventional Diesel design and the originator of hundreds of patents. He had spent a lifetime on the development of the conventional Diesel engines before bringing out the Hesselman design. The significance of this is that the Hesselman engine represents the attempts of a Diesel designer to overcome problems encountered in high-speed high-pressure Diesel-engines. It is not the attempt of a carburetor-engine designer to make his low-pressure engine handle Diesel fuels. In other words, it was originated and endorsed by a leading member of the old Diesel school as a means of overcoming the problems of the high-pressure engine.

Age and Number of Models.—The Hesselman engine can no longer be called experimental. Our company turned out its first production engine about two years ago. Since then we have developed eight different models, ranging in bore from 3¾ to 6½ in., and ranging in power from 25 to 170 hp.

Speed.—These engines have been used for industrial service and are, therefore, of relatively low speed. However, there appears to be no serious handicap in the speed possibilities as there are engines which have operated at 3000 r.p.m.

Brake Mean Effective Pressure.—In the race for brake mean effective pressures, these engines are in the front ranks. A pressure of 104 lb. per sq. in. has been obtained on a production Waukesha Hesselman industrial engine of 6½-in. bore, and laboratory models have shown appreciably higher outputs than this.

Economy.—Production engines will show a full-load economy lower than 0.58 lb. per b.hp-hr. and three quarter-load economy lower than 0.52 lb. per b.hp-hr. In certain

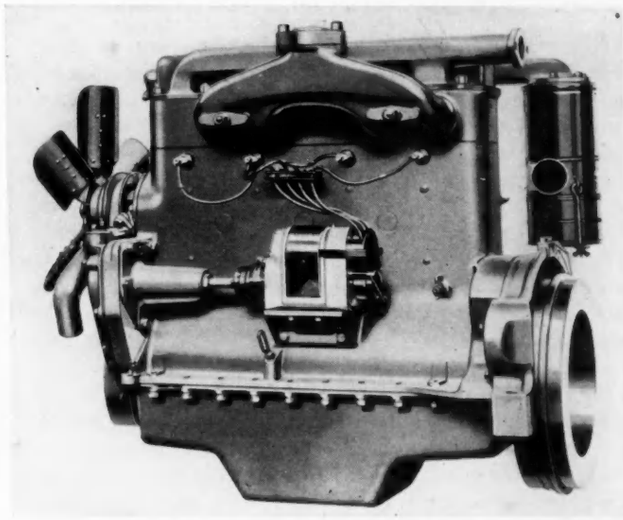


Fig. 1—Pump Side of the Waukesha Hesselman Engine

[This paper was presented at the April 3, 1934, meeting of the Chicago Section.]

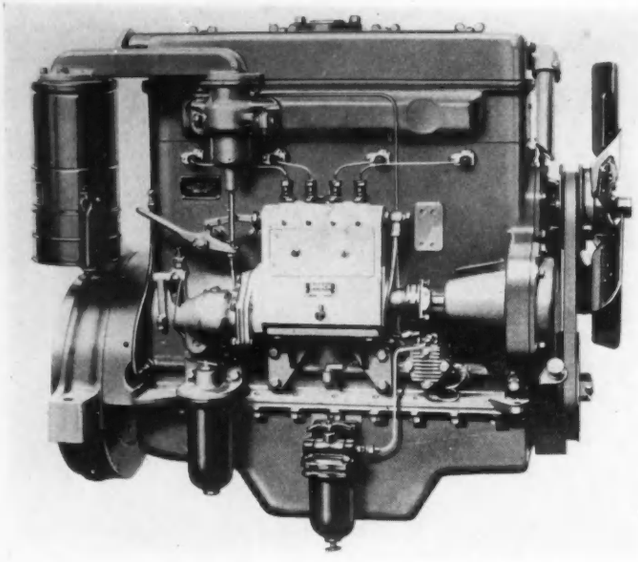


Fig. 2—Spark-Plug Side of the Waukesha Hesselman Engine

cases as much as 10 per cent lower consumption than the above is obtained, and laboratory tests have been obtained with an economy of 0.45 lb. per b.hp-hr. The best general summary of the economy situation seems to be that it lies about half way between the carburetor engine and the Diesel engine.

Cost.—The present cost of the engine is high because of relatively low production. In the smaller sizes, the injection equipment alone costs more than the rest of the engine. It is difficult to estimate what the ultimate costs may be but it seems probable that, with reasonable production, the costs will adjust themselves so that it is profitable to own a Waukesha Hesselman engine if it is used more than 1000 hr. per year, even when the fuel oil is figured at the same price per gallon as gasoline. The present high cost of the engines is very easily absorbed by the artificial and temporary wide spread in the price of fuel oil and gasoline.

Engine Operation Described

Operating Cycle.—Fig. 1 shows the pump side of the engine; Fig. 2, the spark-plug side; and Fig. 3, a section through the cylinder. The operating cycle is as follows: Air is drawn into the cylinder and given a rotary motion as a result of its tangential direction of entry. It is compressed on the "up" stroke of the piston without stopping its rotary movement. At about 50 deg. before top dead-center, fuel is injected toward the spark plug from the opposite side of the combustion chamber. The rotary movement of the air serves two functions: (*a*) to mix air and fuel; (*b*) to carry the mixture past the spark plug where the spark occurs at about 15 deg. before top dead-center. The fuel ignites, and the operation from then on is similar to that of the Otto cycle.

Fig. 4 shows the details of the simple open-type injector. There are three check valves in series to prevent dribbling and combustion pressure backing up into the pump line. There are two spray holes directed at about 30 deg. on each side of the center line of the cylinder. One spray goes against the air swirl, and one spray goes with the swirl. Atomization from the jet is accomplished by means of a rotary swirl given to the fuel just before it enters the spray hole. This is one of the simplest methods of producing atomization without resorting to very small holes. The Hesselman nozzle holes range in size from 0.015 to 0.025 in. There is a small cylindrical chamber just ahead of the spray hole; the fuel enters this through a tangential passage and thus sets up a rotary swirl. This is the method used for atomization in most garden sprays. Dispersion, drop size and penetration, can be controlled by the relative dimensions of the cylindrical chamber and spray hole.

Fig. 5 shows the proportions of the latest high-output chamber. This engine will operate at 3000 r.p.m. and has shown a maximum brake mean effective pressure well above 100 lb. per sq. in. The piston is very little longer than is a conventional piston. The piston rings are located in the walls of the cup, thus providing cooling means for this part of the chamber and permitting higher compression ratios with resulting high output. Fig. 5 shows how well adapted the Hesselman engine is for the conversion of a carburetor-type to a Hesselman injection-type, which can be accomplished by the mere substitution of pistons and cylinder head. The Waukesha Motor Co. is in production on such a conversion for one tractor firm, and others are considering it.

Engine Starts on Gasoline

Starting.—The slogan: "The easiest starting engine in the world," could be used as an advertising slogan without deviating far from the truth. The reason lies in the fact that the engine is started on gasoline supplied to the manifold by a few strokes from a conventional primer pump. There are several factors contributing further to ease of starting. As such small quantities of gasoline are required for the start,

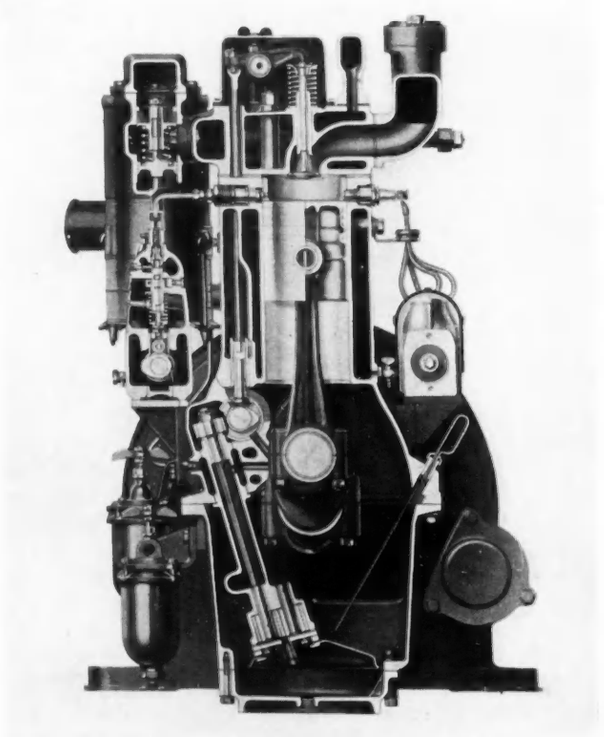


Fig. 3—Section Through the Cylinder of the Waukesha Hesselman Engine

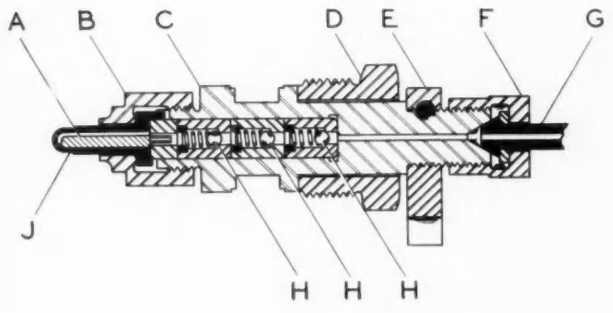


Fig. 4—Details of the Simple Open-Type Injector

it is practical for the operator to keep the primer tank filled with any sort of fuel that is most suitable for starting. A fuel far more volatile than that which would be practical for use in carburetors can be used. The overhead-valve construction causes the fuel to flow into the cylinder at hand-cranking speeds. The cupped piston-head collects and holds any surplus liquid gasoline which is not atomized or vaporized. The compression ratio, which is somewhat higher than are the compression ratios of similar carburetor engines, aids in supplying sufficient heat for a combustible mixture at the spark plug. All of these factors contribute toward making it practical to hand-start any engine that it is possible to pull over compression.

Suitable Fuels Cited

Fuels.—Hesselman engines operate satisfactorily on the standard No. 2 or No. 3 furnace oils. There are many other fuels which operate satisfactorily in the engine, but there are two general limitations in their specification: (*a*) the viscosity must be high enough to provide pump lubrication and low enough to flow readily through filters and to atomize suitably when injected; (*b*) the fuel must not carry suspended asphalts, tars or other products, in quantities sufficient to cause fouling of spark plugs and gumming of valve stems or other engine parts.

Compression ratios are now being used in production which allow the engines to operate without excessive detonation on most fuels encountered. Only one case has been reported where it was necessary to lower the compression ratio. This was an engine operating on kerosene. As kerosene appears to be about the worst knocking commercial fuel and as it is generally more expensive than are the other fuels, it appears to be unnecessary to specify the knock quality or the ignition characteristic of the fuel to be used. This is in marked contrast to the Diesel fuel situation. It is now generally known that fuels which ignite easily are good in a Diesel engine and bad in a carburetor engine or Hesselman engine.

Fig. 6 shows the range of ignition quality covered by fuels of various types. This provides a rather fortunate solution of the fûel problem by simply sorting the low-ignition fuels for Diesel use and leaving the others for carburetor or Hesselman engines. However, this policy is not yet in operation, and there are fuels on the market of such high ignition temperature that they cause serious trouble in the conventional Diesel engine. A Diesel-engine operator, therefore, must know that the fuel he uses is within limits in regard to its ignition property. This is not necessary in the Hesselman engine. However, when fuels are labeled with their ignition property as gasolines are with their octane number, it will be practical to raise the compression ratio of the Hesselman engine with resulting gains in economy and power.

Detonation.—The Hesselman engine is less sensitive to the detonation of fuels than are carburetor engines. The engine is less critical because the fuel is in the cylinder for only a few degrees of crank angle before ignition and is thus very largely in a liquid state in the space ahead of the flame travel. Thus, the heat of vaporization is available for suppressing detonation. Preignition tendencies of the fuel can be eliminated, as it is not necessary to admit fuel to the cylinder until shortly before the proper time for ignition. In general, the Hesselman engine seems to be able to carry from one to two ratios more than can an equivalent carburetor engine.

Light-Viscosity Lubrication

Lubricating Oil.—It is the policy to use lighter-viscosity oils in the Hesselman engines than in corresponding carburetor engines. This is the result of two factors: (*a*) the general recognition of the advantages resulting from lighter oils in respect to gumming and carbon formation; (*b*) the well-known tendency of all injection engines to contaminate the lubricating oil with carbon in a state so fine that it appears in colloidal suspension in the crankcase oil. The lubricating properties of the lighter oils are not seriously affected by the crankcase dilution because, while it may occur in

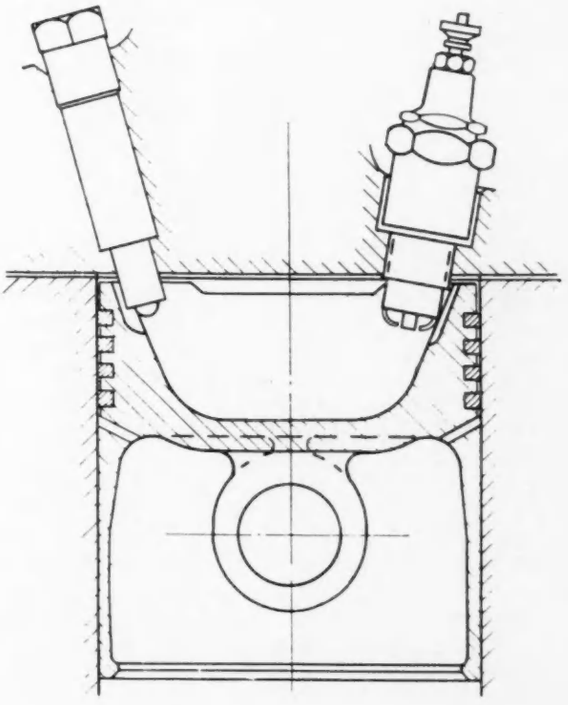


Fig. 5—Proportions of the Latest High-Output Combustion-Chamber

moderate amounts under certain conditions of operation, the lubricating properties of the diluent itself are relatively good as compared with that obtained on carburetor-type kerosene-engines. However, contamination of the crankcase oil is somewhat of a problem, as with all other types of injection engines. There is no filter on the market which will keep the oil clean. The best policy at present is to change crankcase oil at regular intervals. This policy will yield good dividends, even when applied to carburetor engines.

Manifold Vacuum.—An important feature of the Hesselman cycle is the method of throttling the intake air to provide combustible mixtures at the spark plug. From one-quarter to three-quarter load, there is a certain amount of stratification between air and combustible mixture. It is this partial stratification at part load which produces the economy gain over the carburetor engine.

Fig. 7 shows the manifold vacuum plotted against brake mean effective pressure on a carburetor engine and a Hesselman engine. From this it is plain that, at part loads, the air-fuel ratio is definitely higher in the Hesselman engine than in a carburetor engine. On present engines, air throttling is accomplished by mechanical linkage between the fuel pump and the balanced barrel-type throttle-valve. The throttle valve is spring loaded to provide compensation at variable speeds. Production is just starting on a new-type vacuum-control of the fuel pump which eliminates all mechanical linkage. With this the pump setting is controlled by a piston mounted directly on the pump and operated by manifold vacuum. The load on the engine is regulated by throttling the air with a simple butterfly air-valve which corresponds to the throttle valve of a conventional carburetor engine.

Idling.—It is possible to idle engines at a speed as low as 400 r.p.m.; but, for ordinary use, they are not reliable when idled below 600 r.p.m. Under load, however, they may be pulled down to speeds as low as those of a carburetor engine.

Ignition Theory Stated

Ignition.—The ignition theory in a Hesselman engine is rather interesting. It is well known that a conventional spark-ignition system will ignite a gaseous mixture when compressed if it is of suitable air-fuel ratio. However, it would not normally be expected that the spark would ignite the spray of liquid and air in a Hesselman engine, and it would not ignite if there were not one other factor present; namely, residual heat in the combustion chamber sufficient to vaporize a portion of the spray before it reaches the spark point. A cold engine motored over, even at high speed, will not ignite, but a few cycles of operation on gasoline provides sufficient residual heat to accomplish partial vaporization of the fuel at the spark point. This is simply a demonstration of the fact that combustion of a liquid hydrocarbon can not take place with gaseous oxygen until the liquid is also in a gaseous state. This is in contrast to explosives, which are combustible directly from the liquid or solid state, this being possible because all of the elements to the combustion are in the solution, whether it be solid or liquid; whereas, in the case of hydrocarbons, the hydrogen and carbon are in liquid state and the oxygen in the gaseous state.

Indicator Diagram.—Introduction of the Hesselman engine has stimulated reinvestigation of the engine indicator-diagram to obtain a forecast of the probable trend of development. Fig. 8 shows typical pressure-time cards for a 5.8:1 compression ratio, a 9.2:1 ratio and a 14.2:1 ratio. The 5.8:1 ratio represents the present typical Hesselman engine and also the typical carburetor engine. The combustion line is drawn to indicate a rate of pressure rise of 30 lb. per deg. of crank travel. It has been found by experience that, with engines of present proportions, this rate of pressure rise produces practically all the power it is possible to get out of the engine and still provides reasonable smoothness. The rate of change of pressure at the point A where the main pressure rise starts is, of course, an important factor in controlling the mechanical roughness of the engine; but, for the purpose of this analysis, the details of this part of the curve need not be considered.

Experience has indicated that, if a rate of pressure rise greater than 30 lb. per deg. of crank travel is obtained, the tendency to stimulate detonation more than offsets the gain in power. Therefore, we establish the 30 lb. per deg. pressure-rise as the basic ideal-combustion line. The 5.8:1 engine will burn up this line from about 125 lb. per sq. in. compression pressure to a peak combustion pressure of about 490 lb. per sq. in. It then expands down the expansion line to about 45 lb. per sq. in., when the exhaust valve opens. This expansion line will approximately follow the equation $PV^{1.3} = C$. A more accurate method of constructing the expansion line has been proposed by J. C. Slonneger in a recent paper[1]. This consists of taking the compression and expansion line from an indicator of the engine while being motored hot, but not firing. The expansion line on this card takes into account all engine-radiation characteristics, and it

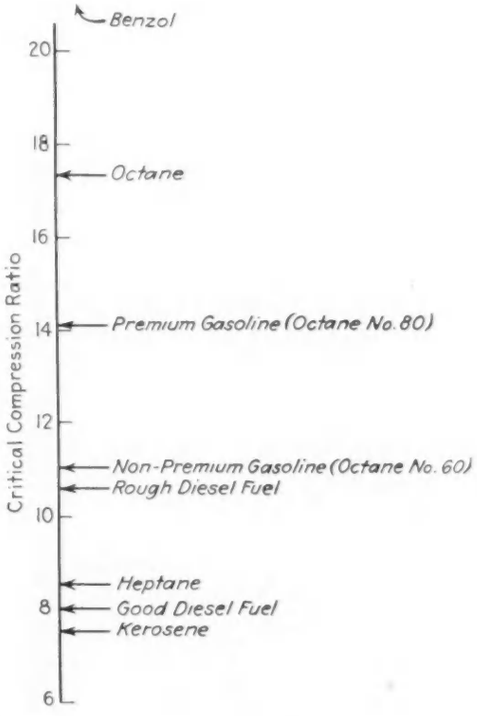


Fig. 6—Range of Ignition Quality Covered by Fuels of Various Types

[1] See S.A.E. JOURNAL, August, 1934, p. 288: Effective Combustion as Determined from the Indicator Diagram, by J. C. Slonneger.

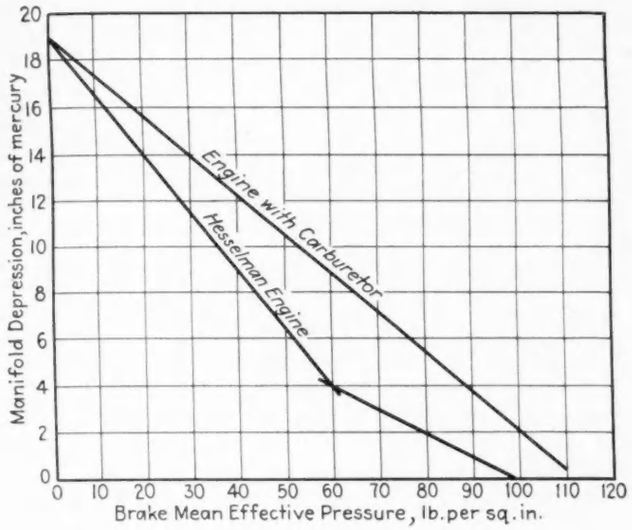

Manifold Depression, inches of mercury
Brake Mean Effective Pressure, lb. per sq. in.
Engine with Carburetor
Hesselman Engine

Fig. 7—Curve Showing Manifold Vacuum versus Brake Mean Effective Pressure for a Carburetor Engine and for a Waukesha Hesselman Engine

is argued that an inert gas expanded from some higher pressure in that cylinder will follow a similar curve. Therefore, multiplying the ordinates of the original expansion curve by any constant will construct the correct expansion curve for an inert gas in that cylinder from any corresponding higher pressure. The point at which combustion has been completed is readily determined by noting when the actual expansion-line becomes parallel to the inert-gas expansion-line constructed through the peak pressure.

The important point to keep in mind on this 5.8:1 indicator card is the fact that practically all of the combustion is completed at 18 deg. after top dead-center. Actual cards from engines indicate that the tip of the diagram is broken off and that the expansion line does not parallel the inert-gas line until 5 to 10 deg. later, indicating about 90 per cent burned at 18 deg. with slight afterburning for 5 to 10 deg. more; but, in order not to confuse the point of this analysis, we consider the combustion as completed at 18 deg. after top dead-center.

Practical Combustion Pressures

General experience with engines of the present design-proportions seems to indicate that it is not practical to carry a higher combustion pressure than 650 lb. per sq. in. Therefore, a dotted line has been drawn at the 650-lb. pressure-level to indicate the maximum permissible pressure in the future ultimate engine. The 30 lb. per deg. combustion line has been extended until it meets the 650-lb. pressure-limit at *H*, which happens at 22 deg. after top dead-center.

To find what compression ratio an engine would have to attain this pressure, an inert-gas expansion-curve was constructed by trial and error, which ran through *H*. This computation was based on the assumption that $PV^{1.3} = C$ and that the quantity of heat available during combustion would be the same as in the 5.8:1 ratio cylinder. This construction indicated that the cylinder would have a 9.2:1 compression ratio and would produce a compression pressure of 240 lb. per sq. in. and that, if it burned along the same 30-lb. pressure-rise line as did the 5.8:1 ratio cylinder, it would reach a peak pressure of 650 lb. per sq. in. and combustion would be completed at 22 deg. after top dead-center.

The only way this cylinder could be made to exceed 650 lb. per sq. in. peak pressure would be to burn the fuel earlier when the pressure might fall anywhere under the line *EH*. However, as the limiting desirable peak pressure was previously established as 650 lb. per sq. in., it would be necessary that this cylinder be designed in such a way that it could not burn faster than along the 30 lb. per deg. line. This diagram would appear to be the ideal toward which future developments will lead, as it burns the fuel as rapidly as practical to as high a pressure as practical with the minimum amount of negative work expended on the compression side of the diagram.

Diesel-Engine Expansion-Curves

It is interesting to carry the analysis still further and plot the expansion curves for a Diesel engine of 14.2:1 compression ratio with 450 lb. per sq. in. compression pressure. The need for rigid control of combustion in a Diesel becomes apparent immediately, as it would be a simple matter to develop pressures well over 1000 lb. per sq. in. if combustion were not under accurate control. It is plain that, in a Diesel engine, we are playing with dynamite, as early combustion would cause excessive pressures. The most important feature of the Diesel diagram is the fact that, if combustion is completed in less than 30 deg., the pressures must exceed 650 lb. per sq. in. It was possible to complete combustion in the 9.2:1 cylinder and 5.8:1 cylinder in 22 and in 18 deg., respectively, without excessive pressure. The Diesel cylinder requires 36 per cent and 66 per cent more time, respectively. It is not a desirable feature to subject the engine parts to protracted duration of peak pressures and temperatures. In practice, this condition is materially worse than the theoretical diagram, as typical cards indicate the duration of combustion in the Diesel to extend for as much as 50 and 60 deg.

The Fig. 8 chart also indicates the relatively large amount of negative work done by the Diesel during compression. The

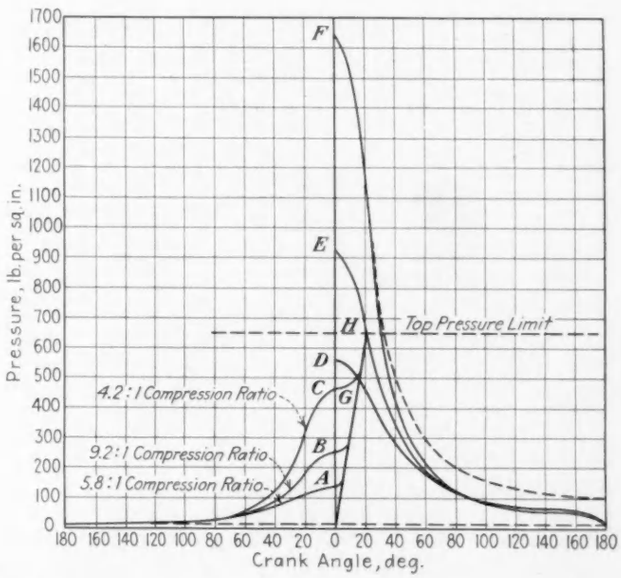


Fig. 8—Typical Time-Pressure Cards for Several Compression Ratios

expansion ratios for the three cylinders taken from the pressure peak to the end of the stroke are 6.9, 6.85, and 5.03 for the 14.2, 9.2, and 5.8 compression ratios, respectively. This emphasizes the possibility of high efficiencies from the Diesels and shows that the high pressures are justified since they result in high expansion-ratios and efficiencies. But the startling thing about this comparison is that the 9.2:1-ratio cylinder has the same expansion ratio as the Diesel and attains the same pressure peak with only about one-third of the excess negative work required by the Diesel cylinder and with very little more burning time than that of the 5.8-ratio cylinder.

The situation can be summarized by stating that the Diesel engine has indicated the practical limits to the use of high compressions. The Hesselman engine has provided a means for handling heavy fuels at low compressions. The future engine should be an engineering compromise, utilizing as many of the benefits of high compression as possible without sacrificing the advantages of the low-compression engine. The diagrams indicate that its compression ratio will fall between 8:1 and 10:1.

Ice Formation in Aircraft-Engine Carburetors

ENGINE stoppage caused by formation of ice in the induction system, particularly in the carburetor, has long been recognized as an occasional cause of airplane disasters. More recently it has caused trouble in automobiles equipped with unconventional carburetors. As this phenomenon occurs only within rather narrow limits of atmospheric conditions, and only with the more volatile fuels or rich mixtures, it has not hitherto been intensively investigated. In December, 1932, the National Bureau of Standards commenced a research on this subject, sponsored by the Cooperative Fuel Research Steering Committee and made possible by a contribution to this committee from the Phillips Petroleum Co.

To have ice formation, it is apparent that we must have a source of water and a source of cooling to 0 deg. cent. or lower. The source of water is obviously atmospheric humidity; that of the cooling is the evaporation of the fuel. A brief consideration shows that ice cannot form unless the mixture temperature is below both the dew point and the freezing point. In consequence, ice formation in the carburetor must depend on at least the following factors: (*a*) volatility and heat of vaporization of the fuel; (*b*) mixture ratio; (*c*) humidity, pressure, and temperature of the intake air; and (*d*) heat transfer between the carburetor and its surroundings, especially the engine.

The difficulties involved in eliminating or correcting for item (*d*) in full-scale tests made it desirable that the effects of the other factors should be studied before going to engine tests. The initial tests were therefore made on a small-scale setup, consisting essentially of a carburetor and means for conditioning the air and fuel supplied to it. The subsequent engine tests were made in the Altitude Laboratory using a Curtiss D-12 engine.

Fuels Used.—For the purpose of the tests, seven fuels of the various volatility characteristics shown in the ccompanying tabulation were used. Fuels 1, 2 and 3 represent the aviation natural-gasoline type. Fuel 1 is a mixture of pentanes and hexanes of volatility beyond the range of commercial gasolines; Fuel 2 represents the extreme volatility in commercial aviation gasolines; Fuel 3 is an admixture with Fuel 2 of a heavier refinery gasoline so that it would comply with the current Army specification which requires that the sum of the 10, 50 and 90-per cent points exceed 564 deg. fahr. (260 deg. cent.). The lettered fuels, *A, B, C* and *D*, are all typical refinery products. Fuel *D* approximately meets the minimum volatility requirements of fighting-grade aviation-gasoline, while Fuels *A, B* and *C* are all commercial-aviation gasolines of higher volatility than Fuel *D*, differing from one another principally in that their 90-per cent points are successively higher in the order named. The tests were made with two separate setups, one small-scale and one full-scale.

Volatility Characteristics

Fuel	*A*	*B*	*C*	*D*	1	2	3
Reid Vapor Pressure, Lb. per Sq. In.	6.0	6.5	6.5	6.5	7.0	7.0	6.5
Percentage Evaporated:	Distillation Temperature, Deg. Cent.						
10	70	70	66	66	61	59	56
50	91	97	96	102	66	73	78
90	109	119	128	135	69	95	128

Conclusions

(1) It can be predicted with reasonable certainty that ice will have formed in the induction system in sufficient amounts to effect an appreciable loss in engine power, or that this condition will be impending, at a temperature of the carburetor venturis about 5 centigrade degrees below the dew point of the atmosphere, within the range of humidities likely to be encountered in flight, when the venturi temperature is at or below the freezing point. The findings with the simulated carburetor setup—small-scale tests—approximate those with the engine, the difference being that the mean line of ice formation appears to be at least 1 centigrade degree higher in the former case than in the latter. Ice disappeared from the carburetor venturi in the small-scale tests at temperatures quite close to the dew point. The temperature at which ice disappeared could not be ascertained definitely in the engine tests.

(2) The small-scale experiments have shown that the venturi temperature in an unheated carburetor can be predicted with considerable accuracy in the service range when the mixture ratio supplied and the distillation characteristics of the fuel are known. To make similar predictions for the carburetor on an aircraft engine, the heat input to the carburetor must also be known or estimated.

(3) For any particular engine operating under specified atmospheric conditions, the danger of ice formation tends to increase both with increasing richness of the mixture supplied by the carburetor and with increasing volatility of the fuel. As noted in Conclusion (1), venturi temperature and air humidity appear to define the danger zone.

(4) The direct application of heat to control venturi temperature is a much more effective means of preventing ice formation than intake-air heating and requires only a fraction of the energy input. Moreover, direct application of heat seems to have little adverse effect on engine power.

(5) The A.S.T.M. corrected 50-per cent temperature is shown to be a much more suitable index than the sum of the 10, 50, and 90-per cent temperature.

—Excerpts from a 1934 Annual Meeting paper by H. H. Allen, G. C. Rodgers and D. B. Brooks, National Bureau of Standards.